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THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

JULY-DECEMBER 1913

INDEX TO THE JOURNAL, JULY-DECEMBER 1913

NOTE: Items appearing in The Journal as Society Affairs have the month and page number given. Where the list of discussors of a paper is given, the list appears under the straight title of the paper. Papers published in The Journal previous to June, discussion of which was published during July to December, are indicated by the month, followed by 1913.

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THE FIRE HAZARD IN TURBO-GENERATORS

By G. S. Lawler

ABSTRACT OF PAPER

There is great chance that serious fire damage will follow arcing in turbo-generators, due to the large amount of exposed, concentrated, combustible insulation, the large amount of energy usually involved in a short circuit, the large quantity of oxygen supplied with the cooling air and the inaccessibility of the fire.

In addition to the ordinary causes, arcing may occur in turbo-generators due to distortion of stator coils and injury to insulation caused by heavy rushes of current unless reactance is provided to limit the amount of current. Generators provided with sufficient reactance may not give trouble until near the end of their life, when short circuits from ordinary causes may be expected. Most generators are not discarded until after at least one short circuit has occurred.

To reduce the chance of fire damage, it is suggested that exposed coils have non-combustible outer covering, the cooling air be filtered and automatically cut off at time of fire and that at the same time carbon dioxide gas be introduced.

On account of their value and the serious results sometimes occurring when their service is interrupted, it is important that turbo-generators be protected from fire.

THE FIRE HAZARD IN TURBO-GENERATORS

By G. S. Lawler,[1] Boston, Mass.

Non-Member

The chances of electric generators of the older types being seriously injured by fire in the event of some part of the insulation failing is slight. Occasionally arcing will ignite the insulation at some point, but it is seldom that the fire will spread much before it is extinguished. This freedom from fire damage is due principally to the comparatively low speeds, the accessibility of the combustible insulation, and the fact that the machines being of large mass per unit capacity, the insulation is considerably distributed.

2 This condition of practical freedom from fire is reversed in the case of generators of the turbo type, for when a short circuit occurs in one of them there is a great chance that the insulation will be ignited and the machine be badly damaged; in fact such damage has occurred in a number of instances.

3 The chief causes of the increased hazard in the more modern type generators are as follows:

a The volume occupied by this type of machine is very much less for the same capacity than that of the older types of generators, so that the combustible insulation is more concentrated and, therefore, much of it is exposed, even to a slight arc or fire. The covering on the conductors depends greatly for its insulating qualities on the presence of oils or gums of a highly combustible nature. The amount of this combustible insulation on the higher voltage generators is naturally greater than in the low voltage machines.

Owing to turbo-generators having only a few poles the end connections between slots form a large proportion of the total length of conductors, in fact in some

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The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

designs approximately one-half of the coils are outside of the slots. These end connections, one-half being on one side of the machine and one-half on the other, are exposed to fire, and as with a pile of loosely laid sticks, fire will rapidly extend from the insulation on one coil to that on the others.

b Owing to these generators being of exceedingly large capacity in many instances, (one of 30,000 kva. capacity now being constructed) an enormous amount of energy is involved in a short circuit, especially at the instant the short occurs and as the arc is confined in the limited space with the combustible insulation, it would seem impossible for the insulation to escape being set on fire at many points simultaneously.

c The machines are cooled by forcing large quantities of air through the spaces between the conductors. The large and constantly renewed supply of oxygen will hasten combustion when it is once started.

The air is given somewhat of a rotary motion by the rapidly revolving rotor which has the ventilating vanes on it and consequently fire when started will be quickly swept around the exposed insulation.

d The generators are totally encased with the exception of the air inlets and outlets and even these in some designs are under the machines. This construction prevents access to a fire and much valuable time will necessarily be consumed before extinguishing agents can be used effectively. When the field current is cut off, as is necessary in case of short circuit, the only means of bringing the rotor quickly to rest is lost and it will continue to run for a long time after the steam has been shut off. Some machines will run for over an hour. This continued rotation is not conducive to the quick extinguishing of fire, especially when the ventilating vanes are mounted on the rotor.

4 In addition to the possible causes of arcing existing in the case of the older types of generators, the turbo-generator is subject to momentary large current rush at instant of short circuit, even if the short is external to the machine itself, unless means are taken to keep the current within safe limits. The heavy rush of current causes mechanical stresses in the conductors, which in

some cases are severe enough to distort the conductors, especially where outside the slots, and to injure the insulating covering, resulting in a short circuit within the generator itself. In some designs the internal reactance of the machines will permit of the momentary current rush amounting to 40, or possibly more, times the normal full-load current of the machines.

5 The possibility of the conductors being distorted has been reduced in some cases by designing generators with sufficient internal reactance, or by providing external reactance such that the current at the moment of short circuit will not be great enough to damage the generators. Attention has also been given to supporting the stator end connections to prevent their distortion. These means have undoubtedly greatly increased the safety of the turbo-type of generator from possibilities of internal short circuit, but in no way tend to prevent a fire resulting should an arc occur.

6 A short circuit in the rotor will probably not result in a severe fire unless under exceptional conditions. This is also true if the short circuit occurs inside of a stator slot. A short circuit involving a stator coil, however, is more apt to occur at the end of the slot where the conductors are exposed.

7 As asbestos is now used largely for insulating the rotor windings and as these windings are well protected, it is probable that only in cases of severe fire in a machine will the rotor windings be damaged to any extent.

8 While the generators may be free from fires during the earlier portions of their life owing to the proper use of reactances which prevent external troubles seriously affecting the machines, as they get older the ordinary causes of breakdown of insulation are liable to occur and fires result. Probably in most cases generators will not be discarded until some trouble, usually in the nature of a short circuit, has occurred at least once in each, so that it is reasonable to expect that unless further preventative means are taken, turbo-generators stand a good chance of serious damage by fire at some time during their life. Although many fires have occurred, probably most of them have happened during the generator development stage. Generators of the turbo type are of such recent production that none of them has yet reached a life which could be considered old and, therefore, the troubles which can be expected near the end of their life by fire have still to come.

9 Undoubtedly the manufacturing companies have given serious thought to the matter of the reduction of the fire hazard in turbo-generators and have employed all means practical at the present time to this end, but there is still very much to be desired. The following several means if taken together would seem to minimize the chances of a serious fire:

a If a suitable material could be found a non-combustible outer covering could be placed over the insulation on the stator end connections. This would greatly delay the spread of fire and even if no other protective means were taken, would undoubtedly prevent much serious damage. Where fire extinguishers were used the covering would at least hold back the fire until they could be brought into play. At present no material suitable for such a covering appears to be available.

b If a non-combustible outer covering should be put on, its advantages would be partially lost in time unless the cooling air were freed of the dirt and oily vapor liable to be in it. This could be done by filtering, as has already been advocated several times.

c Means could be provided for cutting off the air supply in case of fire in generators by placing dampers in the inlet ducts designed so as to be normally held open by fusible links. The links could be placed so that they would be quickly fused by the heat and allow the dampers to close automatically. By reducing the oxygen supply to that entering by leakage the action of the fire would be slow.

d Arrangements could be provided for the quick introduction of carbon dioxide gas into the machines. The carbon dioxide could be kept in liquid form and piped through valves, expansion tanks, etc., to the generators. The valves could be arranged to be opened by the closing of the air inlet dampers so that the gas would be automatically introduced into the generators. This gas would be very effective in extinguishing fires inside the machines after the air supply had been cut off.

10 The employment of some efficient method of reducing the fire hazard in generators of the turbo type either along the lines

mentioned or in some other way is important. The value of these generators is great and the damage by fire may amount to a considerable proportion of the first cost. It is probable that the damage is more liable to occur towards the end of the life of the generators, but even then the loss may be large, both directly and indirectly. The large central stations have reserve units so that the increased damage due to fire in one of their generators would probably not affect the continuity of service, but the increased time necessary for repairs may be long and during this time the reserve capacity will be weakened. In the case of industrial plants the longer time needed for repairs might be serious. Many manufacturing concerns who generate their own current depend on only one unit and, therefore, their whole production, or a large part of it, would be affected.

THE CENTRIFUGAL BLOWER FOR HIGH PRESSURES

By Henry F. Schmidt, Published in The Journal for November 1912

ABSTRACT OF PAPER

In this paper the author gives the essential elements of blower design in convenient form for reference. An equation is derived representing the work done in a centrifugal blower and the differences between this type and a piston compressor are discussed. There is a discussion of efficiencies and characteristics of impellers in series, of guide-vane and volute types of blowers and variations of pressure in the case of guide-vane blowers as the volume of discharge increases. The theory of diffusion vanes is examined in detail, and an equation for the loss of kinetic energy derived.

Design of volute blowers is taken up and discussion of the influence of the relation between efficiency curve and discharge rate, as well as between the blower characteristic and efficiency curve in this type of blower; an equation for the velocity conversion efficiency is derived, and the constancy of pressure in a blower having a free vorte mathematically demonstrated. The Westinghouse design of volute blower is used for illustration.

The latter part of the article is devoted to methods of testing and calculation of centrifugal blowers.

DISCUSSION

W. H. Carrier objected to the author's statement that the loss in radial flow is overcome by the use of the volute casing. This thing was tried out some years ago in pumps, but found by means of tests to be of no advantage. As to the loss at the inlets, everything depends on the relative length of the blades, as the author admitted, a fact which the designers of centrifugal machinery have taken advantage of by proportioning the inlets of their vanes properly. The speaker made tests some years ago to determine the proper size of inlets for a given quantity of air

Presented at the Annual Meeting, December 1912, of The American Society of Mechanical Engineers, 29 West 39th Street, New York. All discussion is subject to revision.

and speed. There is a relationship which can be proved both mathematically and experimentally, between air per minute, speed and diameter of the inlet, the diameter of the inlet being determined by the other two. A fan can be very inefficient if the inlet be too large or too small for a given speed and a given quantity of air, and there actually are many such inefficient fans on the market.

C. J. H. Woodbury said that the drawings of the compression of air do not represent what takes place in the blower. They represent the streams of issuing air as though they were streams of water running through a river, whereas it is merely a difference in density which might be represented by lines nearer together. The speaker also called attention to another form of centrifugal blower, in which augmentation of radial efficiency of the pressure is obtained by making the vanes of variable thickness, so as to make the spaces between the blades at the periphery less than at the sides towards the center, or, in other words, applying the principle of a hose nozzle to a blower.

R. H. Rice, in a written discussion, made the following objections to some of Mr. Schmidt's statements as the result of an extended experience in the design and construction of such apparatus running into the installation and operation of many hundreds of sets, as well as an immense amount of experimental and research work conducted under his direction.

In Par. 8 Mr. Schmidt reaches the conclusion that the maximum efficiency of a centrifugal compressor, without conversion of the final velocity of air into pressure, can be only 50 per cent as a maximum, neglecting all losses. This conclusion is correct. It is also correct, as he states in this same paragraph, that there are three means of effecting this conversion. But in Par. 9 he admits that the first method is the most efficient at the designed capacity, while incorrectly stating that this method is inefficient at large and small discharge rates.

In Par. 11 he states that diffusion tubes attached to the discharge are inefficient, but he incorrectly gives the reason for this inefficiency. The diffusion tube is simply the down-stream half of a venturi meter, and we are all familiar with the fact that venturi meters are efficient for measuring both compressible and incompressible fluids. If this portion of the meter were ineffi-

cient as a converter of velocity into pressure, the meter itself would be inefficient. The reason for the inefficiency of a diffusion tube attached to the discharge is the long and tortuous path which the air is required to pass over after leaving the impeller before reaching the diffusion tube; and the fact that this air moves at a very high velocity in so doing.

In Pars. 12 and 13, Mr. Schmidt shows that the efficiency of the centrifugal blower remains the same with a reduced number of revolutions provided the volume is reduced, but does not give the law, which is, that quantity divided by number of revolutions, $\frac{Q}{N}$, must be constant.

In Par. 15, it is stated that the characteristics of a guide vane blower are shown by Fig. 6, and that the pressure fluctuates in such a machine throughout the range of volume. This statement is quite incorrect. The only fluctuations of pressure met with in a guide vane blower are those due to a rising pressure characteristic at the early part of the volume pressure curve; and in a properly designed volute blower the pressure volume curve has the same characteristics and the same fluctuations are met with. In the guide vane compressor when properly designed there are no pressure fluctuations for loads greater than 1/3 or ½ of the rated volume as can be seen by inspection of any of several hundreds of these machines which the writer has designed and which are now in operation all over the country. Furthermore the relation of the pressure and efficiency curves, shown in Fig. 6, are those of an improper design. No designer experienced in the art of guide vane blower construction would find it necessary to place the peak of the efficiency curve at a point where the pressure has dropped to less than half the maximum. Consequently, the conclusions drawn from inspection of Fig. 6 are incorrect both as regards the character of the pressure curve, and as regards the relation of the pressure curve to the efficiency curve.

In general a volute blower will have larger dimensions than a guide vane blower to effect the same result; and the frictional and eddy losses in a volute blower will be greater than those in a guide vane type, due to the imperfect guidance of the air and somewhat greater surface exposed to friction.

In Par. 18, Mr. Schmidt states that no pressure is created by the diffuser type, or, by implication, by the discharge vane machine, when no volume is being delivered. This statement is,

however, not correct. An appreciable increase in pressure over that due to the centrifugal effect is met with in properly designed discharge vane machines.

In Par. 23, Mr. Schmidt states that the volute blower shown in Fig. 5 has an entirely different characteristic, the theoretical characteristic being, namely, a perfectly straight, or constant pressure, line for any delivery. This characteristic is not peculiar to volute blowers and referring to Par. 24, it is shown in Fig. 22 that the characteristics of the efficiency curve are the same in both types of machines, with the exception that the peak of the efficiency curve in the volute machine, shown by Mr. Schmidt, is much lower than the peak of the efficiency curve in a properly designed discharge vane machine for the reasons before set forth.

The real reason for the flat pressure curve of Mr. Schmidt (Figs. 7 and 10) is that he is not getting the proper conversion of velocity into pressure as the load comes on; therefore, he is not realizing from a given diameter of impeller the full amount of pressure possible with good design. In this connection Fig. 23 shows a machine with good velocity-pressure conversion (in full line), and the dotted lines show the result to be obtained if the velocity was not fully converted into pressure in the volute. This figure refers to a machine with volute casing, no discharge vanes, and so-called free vortex.

Pars. 30-34 are devoted to an attempt to show that a radial inlet taking the air without shock is not inefficient. The obvious answer to this contention is Par. 35 in which it is shown that in an attempt to overcome the losses of such impellers, the inlet is made in the form of a volute and forced vortex. No doubt such a construction can be made to increase the efficiency, or decrease the losses of impellers with radial inlets, but it is impossible that this construction should give as satisfactory results as the one in which the edges of the impellers are formed to take the air without shock and without whirling, and extensive experiments show that it does not.

In Par. 37 Mr. Schmidt states that there is only one good construction for impellers. Engineers who are accustomed to deal with the problems of actual design will realize the impossibility of fixing on any one construction as the best for all circumstances. Furthermore, Mr. Schmidt makes it appear to be impossible to construct successfully a multi-stage machine since he states that the disks should not be provided with any hub or hole

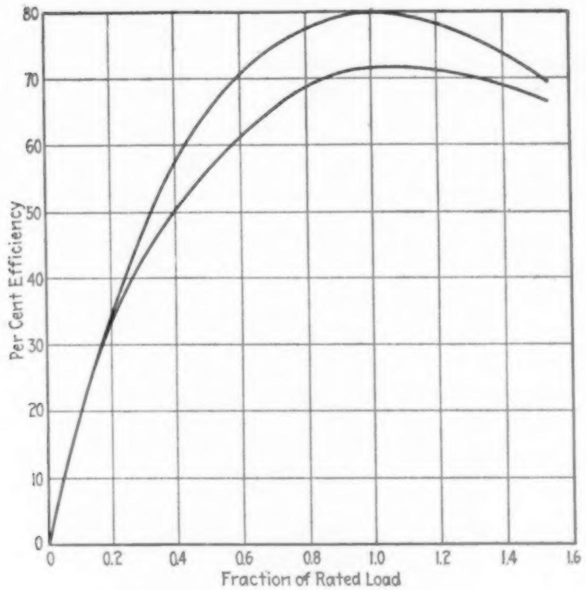

FIG. 22 EFFICIENCY CURVES OF VOLUTE AND DISCHARGE VANE MACHINES

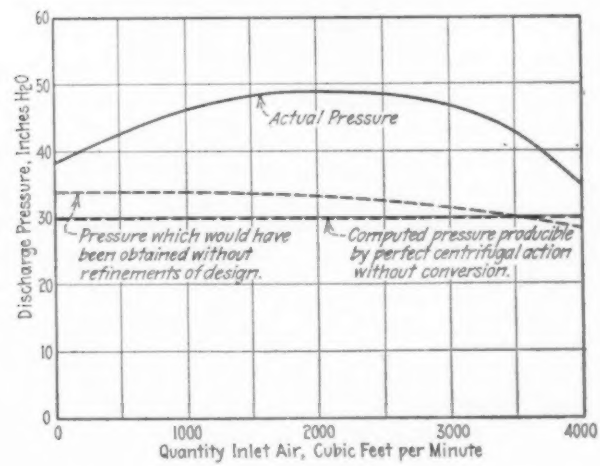

FIG. 23 PRESSURE CHARACTERISTIC OF MACHINE WITH VOLUTE CASING, FREE VORTEX AND NO DISCHARGE VANES

for the shaft. The fact that hundreds of machines are operating with this form of construction apparently has no weight with Mr. Schmidt. This question is safely left to the judgment of engineers as to the possibilities of the situation.

In Par. 36 the author states that his straight radial blades are the only ones which can be used except at the lowest speeds. The impeller illustrated in Fig. 24, which has the blades so arranged that all the metal is disposed along radial lines (while at the same time the air is met without shock), shows that there are

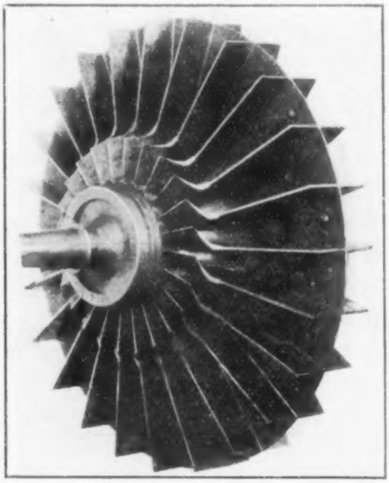

Fig. 24 Impeller with Metal Arranged along Radial Lines

constructions not mentioned by Mr. Schmidt, which are adequate for all situations.

It will be noted in the impeller shown that all the vanes extend to and derive support from the hub, thus gaining strength not possessed by Mr. Schmidt's construction.

S. A. Moss (written). Mr. Schmidt assumes that what he calls volute blowers are inherently more efficient at light loads than what he calls discharge vane blowers. However, the data which he gives in Fig. 10 show quite the contrary, as Mr. Crissey has independently noticed. The blower which he marks 40,000 cu. ft. has a curve with rising pressure characteristic and this gives higher efficiency than the curve which he marks 30,000 cu. ft., which has no pressure rise above no load. I have replotted Fig.

10 as Fig. 25, using fractions of best load as abscissae. The best load for the 30,000-cu. ft. curve is at 25,000 ft. and all of the abscissae of this curve have been divided by this number. The peak of the 40,000-ft. curve is not shown and has been assumed 60,000 and all of its abscissae have been divided by this number. This brings the two curves to a common basis. As will be seen the machine with the rising pressure characteristic has a higher efficiency at the peak and rises more rapidly at light loads. Mr. Schmidt states in Par. 24 that curves such as Figs. 6 and 7, if plotted to scale, would show decreased light load efficiency for guide vanes. This is not borne out by the facts he presents however.

A blower of the type discussed, with a rising pressure characteristic, gives such rise of pressure purely by better velocity conversion with the same shaft power input, and hence necessarily better efficiency than a blower which gives no pressure rise above the no-load value. As Mr. Rice has mentioned this rise can be secured by discharge vanes or a good volute.

Mr. Schmidt introduces in Pars. 25 and 26 a discussion of the mathematician's free vortex. The presumption is that he uses this in his blowers as well as the volute casing of Fig. 5. He attempts to show that a blower with a free vortex would give a flat characteristic curve with considerable velocity conversion at all loads. At very light loads, however, there is enormous friction loss in a free vortex since the angle between the path of the fluid and a tangent is very small. At a very small load close to zero, a particle of fluid must travel around the circumference many times before it gets out. At slightly larger loads the particle travels around fewer times. As the load increases the path of the particle becomes shorter and shorter, and the friction and eddy losses become much smaller percentages. Hence one would expect a volute with a free vortex to give a curve such as was actually obtained in Fig. 23, presented by Mr. Rice, with increased conversion as we go away from zero load.

Hence a free vortex cannot give good conversion at very light loads on account of the large proportion of friction and eddy losses. If now the pressure does not rise at full load, it follows that undue friction losses exist there also, which can however be removed by efficient design. Hence an actual blower which gives a flat characteristic does not do it because of inherent design, but

because the upper portions of pressure and efficiency curves are lost.

Mr. Schmidt gives a discussion of the action of a diffuser tube in connection with Fig. 8, in which he rightly states that for each velocity at the point of minimum section there is a pressure at the end of the tube which gives equilibrium. Mr. Schmidt's own explanation shows that there is no fluctuation of pressure in a blower using an equivalent of a diffuser tube. As the vol-

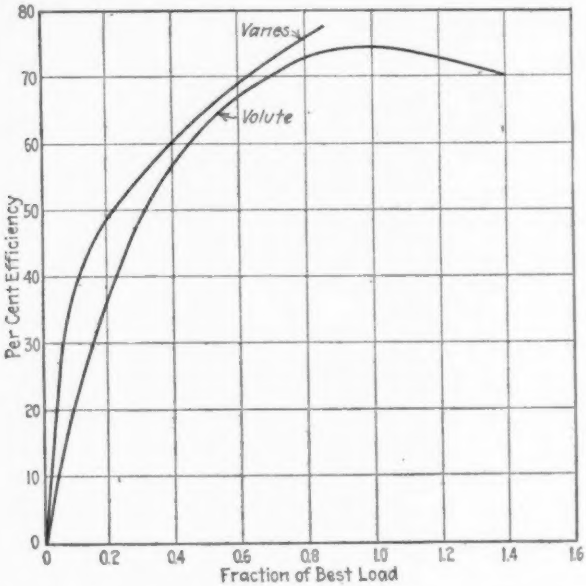

Fig. 25 Characteristics of Two Blowers under Different Efficiency Curves

ume passing through the blower changes so as to change the velocity at the beginning of the tube, the blower itself produces a pressure corresponding to the new conditions. In other words, for every load or volume passing through the blower there is a certain pressure which the blower itself produces due to the particular velocity in the conversion tube and to all the other occurrences when this volume is passing. That is to say, there is a perfectly definite pressure volume curve such as shown in upper part of Mr. Schmidt's Fig. 10 and for any given volume which is being used the pressure produced by the blower is the pressure occurring at this volume on the pressure volume curve. This is the pressure which exists in the pipe line to which the blower

is connected. If the load and hence the velocity is changing from one value to another, the pressure gradually changes to the new value without any fluctuation or pulsation. Mr. Schmidt supposes the case of a high velocity jet discharged into a receiver through a diffuser or venturi tube. It is, of course, to be understood that there is an orifice leading from the receiver so that the air entering can also leave and that there is no other air supply to the receiver. If now the velocity at the entrance to the venturi tube be gradually decreased there will, as Mr. Schmidt supposes, be a new pressure of equilibrium and the pressure in the receiver will gradually and calmly vary until it is again in equilibrium with the velocity. The explanation which Mr. Schmidt gives shows that there will not be any pulsation. The pressure in the receiver is fixed only by the velocity at the entrance to the venturi tube and is always the pressure in equilibrium with this velocity. There are thousands of venturi meters in use which illustrate the same point. These make use of a diffuser tube such as Mr. Schmidt mentions and discharge into a pipe system beyond corresponding to a receiver. If now the flow through the system is reduced, the velocity at the throat of the venturi meter at entrance to the venturi tube decreases. This decreases the pressure rise along the venturi tube, but in a slow and gradual manner and a new position of equilibrium is reached without any fluctuation or pulsation whatever. Evidently the same action takes place in any type of diffuser or volute or free vortex which accomplishes conversion of velocity into pressure.

The mathematical portion of Mr. Schmidt's paper can be given in the following simplified way. In the first place we note that the theoretical work for adiabatic compression of 1 lb. of air from absolute pressure P_2 to absolute pressure P_1 in ft-lb. is

$$\frac{\gamma}{\gamma-1}\,\frac{144\,P_o}{\rho_o T_o}\,T_2 K$$

where

γ = ratio of specific heats

$K = \left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} - 1$

T_2 = initial absolute temperature

P_o, T_o, ρ_o = absolute pressure in lb. per sq. in., absolute fahr. temperature; and density in lb. per cu. ft., at any standard condition

Now consider a pound of air which enters a rotating wheel at the center and passes outward to the periphery always with negligible velocity. Work is done against the forces which keep the relative radial velocity negligible and this work appears as work of adiabatic compression of the air. The force at any radius, r feet, against which the work is being done, is the so-called centrifugal force, which for a pound of air is well known to be

$$\frac{r}{g}\left(\frac{2\pi N}{60}\right)^2$$

This force varies uniformly along the radius so that the average value is one-half the maximum value at the outside radius r_1. The work done is this average value multiplied by the total distance which is the maximum radius, r_1, which is therefore

$$\frac{1}{2g}\left(\frac{2\pi N r_1}{60}\right)^2 \text{ or } \frac{\mu_1^2}{2g}$$

if μ_1 is the peripheral velocity of the wheel in ft. per sec. This is the work done in ft-lb. in passing a pound of air from the center of a rotating wheel to the pressure at the outside of the wheel, the relative velocity along the wheel always being negligible. This is necessarily equal to the adiabatic work of compression and delivery to the pressure existing at the periphery of the wheel given above. This covers all of the points given in Mr. Schmidt's Pars. 6 and 7 leading to his equation [2]. It is to be noted that this covers only the case where fluid is taken on the wheel at the mathematical center and where the relative radial velocity along the wheel is always negligible. Extension to the actual case where the fluid is taken on the wheel at a considerable radius and where there is an absolute tangential velocity of the fluid at the point where it is taken on the wheel as in Mr. Schmidt's Fig. 15 and where the relative velocity along the wheel and particularly at the circumference is not negligible but has an appreciable value, are matters which must be analyzed in addition and which Mr. Schmidt wholly omits. Mr. Schmidt's analyses of the actual occurrences in a centrifugal blower are therefore only approximate.

Mr. Schmidt's method of calculation of the theoretical horsepower of a blower given in Pars. 45 and 46 can be simplified by computing independently the quantity discharged from the ori-

fice in cu. ft. of air per min. if reduced to absolute pressure and temperature existing at the blower inlet. This avoids altogether direct computation of the velocity or volume at orifice end conditions. Using the analysis which Mr. Schmidt gives in Par. 46, the volume discharged from the orifice with total pressure P_1 absolute temperature T_1 expressed in cu. ft. per min. if reduced to actual absolute temperature T_2 at inlet and pressure P_2 at orifice end and inlet is

$$Q = 6540 \frac{A T_2}{\sqrt{T_1}} \sqrt{(K+1)K}$$

Here 6540 is $60 \sqrt{2gJC}$, and A is the orifice area in sq. ft. multiplied by the velocity coefficient. K is the same as above if discharge and orifice pressure are both P_1. The theoretical horsepower for compression and delivery per minute of a body of air occupying a volume of one cu. ft. at inlet conditions is

$$H = \frac{144 P_2 K}{33{,}000} \frac{\gamma}{\gamma - 1}$$

Here P_2 is absolute inlet pressure in lb. per sq. in. The theoretical horsepower for the blower is then HQ. This is an expression equivalent to Mr. Schmidt's equation [10], and I believe will be found easier to handle. The expression for Q can also be simplified by means of a very exact approximation originated by Bossinesq and discussed in a paper on the general subject of discharge of air in American Machinist.[1]

C. P. Crissey (written) expressed his belief that the author has compared the Westinghouse volute blower with turbo-blowers of inferior design, which makes the conclusions inadmissible. Neither pulsations over the range of volume indicated in Fig. 6, nor sharp peaked efficiency curves with rapid drops on either side of the peak are inherently characteristic of the turbo-blower, and indicate simply poor design.

The author gives some curves in Fig. 10 but supplies insufficient information regarding them. Even the speed of the turbo-blower is lacking and we are not informed concerning such points as the clearance between impeller and diffusion vanes, whether the vanes are straight and radial, straight without being radial, or curved.

[1] Sept. 20, 27, 1906.

The comparison shown in Fig. 10 might well be plotted with abscissae taken as fractions of the rated capacity. It will be seen that the volute blower reaches its maximum efficiency before full capacity, while the turbo-blower efficiency is still rising beyond one and one-quarter load. This latter curve shows again that the design of the turbo-blower is not correct, because it is perfectly feasible to have the efficiency curve of a turbo-blower reach its maximum at full load or at less than full load quantity as desired. It will be noted that the efficiency curve of the volute blower falls off about 8 per cent from the maximum at half and at one and one-quarter capacity. The best turbo-blowers for such low pressures and large capacities will show less reduction in efficiency for corresponding points, when the efficiency is a maximum near full load, and the maximum efficiency will be higher than that of the volute type; therefore the efficiency throughout the usual working range is better.

Every builder of turbo-pumps, turbo-blowers and turbo-compressors would like to omit diffusion vanes because they add to the expense of production, but experience has shown that they are advantageous. Their use led to the introduction of multi-stage high-pressure centrifugal pumps, and experience with turbo-blowers and compressors confirms their value. After careful experiments, Professor Rateau advised his liensees to construct machines without diffusion vanes, but the results were not as expected, and these builders are now in general using diffusion vanes; even careful experimental researches are not always conclusive.

Par. 36 states that straight radial blades are the only ones which can be used except at the very lowest speeds. The majority of turbo-blowers and compressors in service throughout the world do not, however, have such blades. Par. 37 is also misleading, as the great majority of turbo-blower and compressor manufacturers use with satisfaction impellers of built-up construction, while a number of those who began by using solid impellers have abandoned them. Double flow impellers are very good, but single flow are equally satisfactory and are widely used for multi-stage work upon the continent of Europe because of the complications in design and increase in length, weight and cost resulting from connecting a number of double flow impellers in series. The statement that an impeller disk should not have a hub or hole for the shaft almost refutes itself, as thousands of rotating

disks with holes for shafts are today operating with satisfaction in steam turbines, blowers and compressors.

The writer prefers the European method of test to that used by Mr. Schmidt. It is usual to rate and guarantee these machines in terms of atmospheric temperature and pressure; therefore the use of a nozzle in the discharge necessitates numerous and large corrections all tending to introduce errors, to say nothing of radiation effects which take place at high air temperatures. Experience has shown it to be more accurate as well as simpler to use a nozzle in the suction pipe. The formula for the weight of air entering the compressor is then

$$G = 2.056\, FP_2C \sqrt{\frac{B^{0.286}}{T}(B^{0.286} - 1)}$$

in which

G = weight of air in lb. per second.
F = area of nozzle in sq. in.
P_2 = absolute pressure within suction chamber in lb. per sq. in.
C = nozzle coefficient.
T = absolute temperature of atmosphere in deg. fahr.
P_1 = absolute pressure of atmosphere in lb. per sq. in.
$B = \frac{P_1}{P_2}$

Attention is called to two minor errors. In Par. 13 the work done per lb. of air is said to vary as the number of revolutions. This should of course read as the square of the number of revolutions. The speed for the volute blower of Fig. 10 is given as 3600 r.p.m. on the volume pressure curve, and 3000 r.p.m. on the efficiency curve. Doubtless both curves are for one speed or the other. The symbol P_1 has been used in a number of places to represent the total head (static plus velocity), but in the appendix it is stated to represent only the static head. It is correct in itself, but leaves the significance of the symbol P_2 doubtful.

C. G. de Laval. The notes by the author about centrifugal blowers are interesting and make an attempt to supply the knowledge relating to turbo compressors and blowers of which not much is known up to the present. However, considerable valuable data that may change the theory presented by the author can be found in the able articles relating to this in Dingler's polytechnisches Journal, April 1912, and as early as October 1910,

in the Zeitschrift des Vereines deutscher Ingenieure, Zeitschrift für das Gesamte Turbinenwesen of September 1912, and Theorie and Konstruktion der Turbo-Kompressoren von Ostertag.

The steam turbine is responsible for the introduction of the turbo-compressors and followed rather behind the turbo-pump. The design of modern turbo-blowers follows that of the turbo-pumps and not that of the ordinary volute pumps. Experiments by Rateau and Parsons, Jaeger, Pokorny & Wittekind, and others, distinctly point to the rational design of diffusers as solving this problem.

Parsons, however, in his early machines proceeded on a theory of axial compression, but as soon as he found out the excellent results of Professor Armengaud and others, he produced similar blowers. Without going into the merits and demerits of axial and radial machines it can be said that all modern designs show a radial development for turbo-compressors as entirely opposed to the development of steam turbines, which are almost entirely designed on the axial system.

The statement that the transforming velocity into pressure by three ways with the stress laid upon the method of the use of a volute of proper design is in theory not correct. Only one method used today for transforming this velocity into pressure seems satisfactory, and that is the use of a diffusor or what may be termed a long nozzle, which we all know is the best for liquids. A diffusor is only a series of nozzles of sufficient number to take care of the volume.

There appears to be nothing different in a turbo-compressor than in a turbo-pump except as to the reduction of volume and its increase in temperature; the only difference is the unit of the gas or air pumped which is much smaller than water. The theory applying to a centrifugal or turbine pump can be fully applied here, and the pressure multiplied with the volume being constant is enough to calculate all data in connection with these machines. The relation between pressure and velocity is $p = \frac{v^2 S}{2g}$, in which v = velocity and p = pressure, S = specific gravity and g = acceleration. As air weighs only a fraction of water its ratio to water means that it will take that much greater peripheral velocity to sustain the same head of water. The principal equation to consider in a turbo-compressor is pv^n = con-

stant where $n = 1$ or $n = k$, the former for isothermal and the latter for adiabatic compression.

Without going into the application of these formulæ we may say that the most rational theory applied to impellers is that derived from turbine pumps, with the addition of a shockless entrance and exit to impellers, from which the theoretical head can be calculated for each impeller.

The different purposes for which turbo-blowers or turbo-machines are used require more or less steep characteristics, the same as in turbine pumps. While the characteristics of a pump are practically independent of any change in temperature the turbo-blower is influenced by it to some extent. Characteristics herewith are given for different services and taken from diffusion vane machines (Fig. 26). A constant discharge air pressure for constant speed is no reason for a flat characteristic; on the other hand a steep characteristic curve is desirable when different pressures are to be attained at constant speed. The designer of a diffusion vane machine can obtain any type of characteristic he desires, and a flat characteristic curve is obtained by changing the vane angle. The diffusor or volute has very little influence on the slope of the curve. The efficiency curve should always be as flat as possible. A volute type of blower or compressor is not one that can be adopted for medium and high pressures on account of the difficulty in combining a number of volutes so as to form a unit.

As to the idea advanced as to guide or diffusion vanes and guide vanes, it has been shown by extended experiments that the maximum efficiency has been obtained by diffusors of various shapes and that such vanes do not take away any mechanical efficiency or affect the thermic efficiency on account of friction and heat.

Referring to the method of measurement of capacity which is by the discharge, the guarantees are always based on a certain displacement of air at atmospheric pressure, and it is therefore best to determine the capacity by an orifice attached to the inlet pipes. If the air is measured at the discharge, the kinetic output of energy and radiation losses must be considered and allowance made. The measurement at the entrance is simpler and in the calculation of the results much time can be saved which often extends over days in order to obtain sufficient points for the characteristic.

With the orifice in the air entrance, the correction for temperature or barometer as well as the doubtful determination of the kinetic energy in the output air is dispensed with.

As regards the determination of the adiabatic efficiency from the temperature it may be said that this is purely theoretical and that in practice the radiation losses and the heat caused by bearing friction do not appear in the heat absorbed by the air,

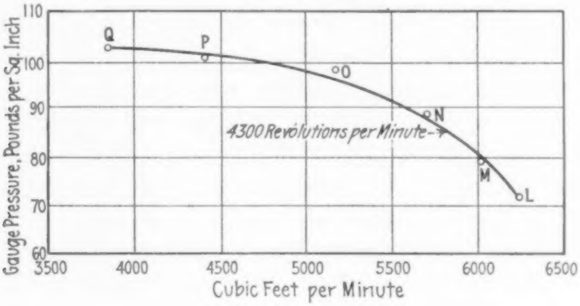

FIG. 26-A CHARACTERISTIC CURVE FOR TURBO COMPRESSOR

the temperature of which must therefore be lower than the theoretical. The formula for efficiency

$$E^1 = \frac{T_1 - T_2}{T_1' - T_2}$$

where

T_1 = theoretical final temperature of the air calculated from the adiabatic relation between pressure and temperature

T_2 = initial temperature of the inlet air

T'_1 = final observed temperature

is therefore imperfect and to the final temperature must be added the above mentioned losses and a third loss caused by the kinetic energy of the air leaving the compressor. This energy converted into pressure would cause an additional rise in temperature.

If we designate the temperature equivalents of the various losses as follows

T_f = bearing friction

T_r = radiation

T_k = kinetic energy

the above formula then becomes

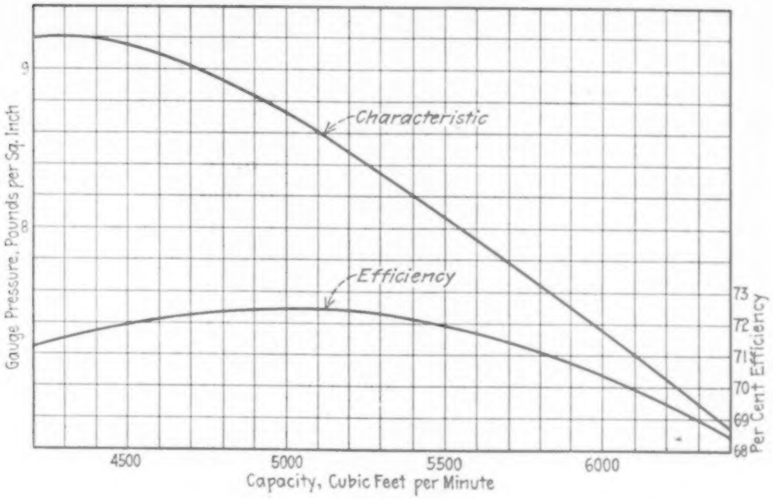

FIG. 26-B CHARACTERISTIC CURVE FOR TURBO BLOWER

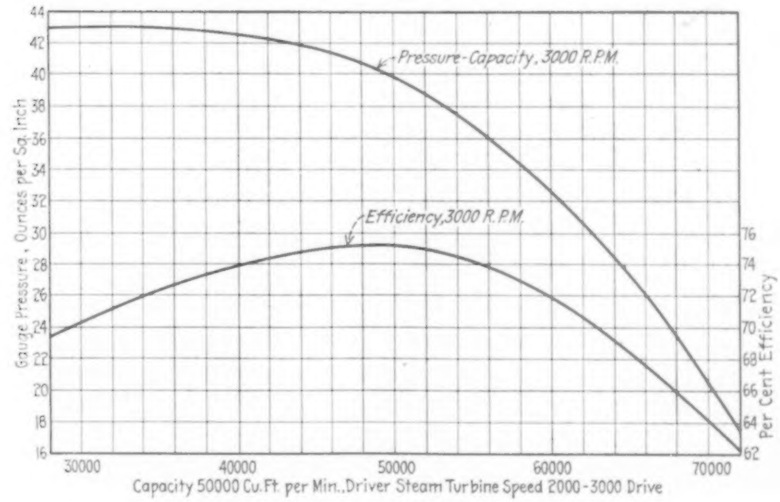

FIG. 26-C CHARACTERISTIC CURVE FOR TURBO BLOWER

$$E' = \frac{T_1 - T_2}{T' + T_f + T_r + T_k - T_2}$$

In the article under consideration the following formula is given:

$$E' = \frac{\frac{\gamma}{\gamma-1} RT_2 \left[\left(\frac{P'}{P_2} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]}{\frac{\gamma}{\gamma-1} RT_2 \left[\left(\frac{P'}{P_2} \right)^{1-\beta} - 1 \right]} = \frac{T_1 - T_2}{T^1 - T_2} \qquad [21$$

which, however, is nothing but the fundamental equation

$$E = \frac{\text{air horsepower}}{\text{power input}}$$

Formula [21] can therefore be used only when the losses are known and serves at best to check the air horsepower or the efficiency in relation to the indicated horsepower.

The laws governing turbo-blowers or compressors are well understood by engineers and this new style of machine which has been brought out will be a factor in the compressor field which is now occupied by displacement machines, the efficiency of turbo-blowers being equivalent to piston blowers working under same conditions. This fact has been rather difficult to prove on account of the different methods used by measuring efficiency of both types.

The judging of a piston machine by figuring efficiency from diagrams obtaining the mechanical efficiency accurately, is of course not right as it includes only the external mechanical losses, and does not include losses due to heating and incomplete filling and expansion of air taken into the valves, all of which will reduce the volumetric efficiency. A piston machine to be compared with a turbo-machine should have the quantity, weight and pressure measured after it leaves the machine and the amount of power used for compressing this quantity without increase in temperature.

Albert E. Guy. Two points of main importance are to be considered in the construction of an otherwise well designed centrifugal air compressor or pump: One is that the direction and the velocity of the fluid at the point of contact with the first element of the vanes, or vane inlet, are fundamental factors in the calculation and delineation of the shape of the vanes. Therefore, when anything occurs which deviates the direction

and changes the velocity of the fluid at the vane inlet, the performance of the machine will be different from, and invariably not so good as was intended. The other is that, as the fluid leaves the impeller nothing can add to the energy it possesses, and everything it contacts with, be it diffuser passage, diffuser vanes, free whirlpool, or easy return bend, will cause a reduction of that energy.

The energy producer is the impeller; it must be designed to meet the conditions imposed by the customer and which may be specified in various ways. No contact for either a centrifugal pump or a blower should be let which does not include a set of guaranteed curves, one of which, named the characteristic or performance curve, represents the variation of head corresponding to the variation of the volume delivered, the speed remaining constant throughout the whole range. The second curve represents the efficiency, that is the ratio between the useful work and the total input work. The third curve represents the brake horsepower. The second and third curves are so placed relatively to the first that both the efficiency and brake horsepower corresponding to one head-volume point on the performance curve can be measured on the ordinate of that point.

The specifications imposed may require that three given sets of conditions be met at the same speed, for instance. The performance curve must therefore enclose three points and yet be a fair curve. It may happen that two of the points cannot be exactly placed on the curve, but once the curve is established to the satisfaction of both the prospective customer and the designer, the impeller is entirely determined. The beginning of the curve at no delivery determines the diameter of the impeller for the stated constant speed, of course, and the post normal part of the curve, that is, the part following the normal point or point of maximum efficiency, determines the areas of sections throughout the impeller. The inlet and outlet angles are fixed by the conditions at the normal point. The shape of the vanes depends on the theory followed.

The apparatus once installed should be tested and the results of the tests embodied in the form of curves. Such curves should then approximate very closely those embodied in the contract. It is quite evident that a wrongly designed impeller cannot produce test curves matching the proposed curves.

Some engineers and writers have claimed that as long as the impeller vane has the proper angles at the inlet and outlet respectively, its shape between these points is of no special importance. This is absolutely wrong. The writer in more than 20 years of practice has found it absolutely necessary, in order to produce a high grade machine capable of meeting predetermined conditions of operation, to calculate the shape of the vanes from the beginning to the end with the greatest degree of accuracy. All the formulae in use today for calculating the impeller can be easily shown to originate from Professor Combes' formula. The three formulae given in Mr. Carl DeLaval's book on Centrifugal Pumps, for instance, are clearly derived from the Fink formula published in the German handbook "Die Hütte," and Fink's formula is clearly derived from Combes'. These formulae have been published in various forms by different authors, but like the thermodynamic formulae at the time of Clausius, they only consider the initial and end conditions, and disregard the work produced between them. However a skillful designer may, by using the Combes formula, produce impellers which will cover quite a range of conditions.

The author's statement, in connection with Figs. 6 and 7, in the writer's opinion are contrary to facts and his theory is obviously erroneous. The compressor described is the same as that which formed the subject of the discussion by R. N. Ehrhart of a paper on turbo-compressors[1] and is simply an enlarged fan of the type first described and made by Professor Rateau some years ago. The high speed at which the impeller runs requires that the vanes be made radial and carved out of the solid, hence shock is unavoidable at the vane inlet, the fluid path throughout the impeller is uncertain and no theory can apply to this sort of design unless a special coefficient of correction be introduced after careful deductions from a large number of accurately made tests of apparatus of various sizes. There is no evidence produced by the authors that such tests have been conducted.

The paper would lead to the belief that the shape of the casing has a great influence on the characteristic curve. This is on par with several patent specifications quoted by Poillon in his book on pumping machinery, in which some French inventors about 30 years ago asserted that the impeller had, of course,

[1] Trans. Am. Soc. M. E., vol. 33, p. 396.

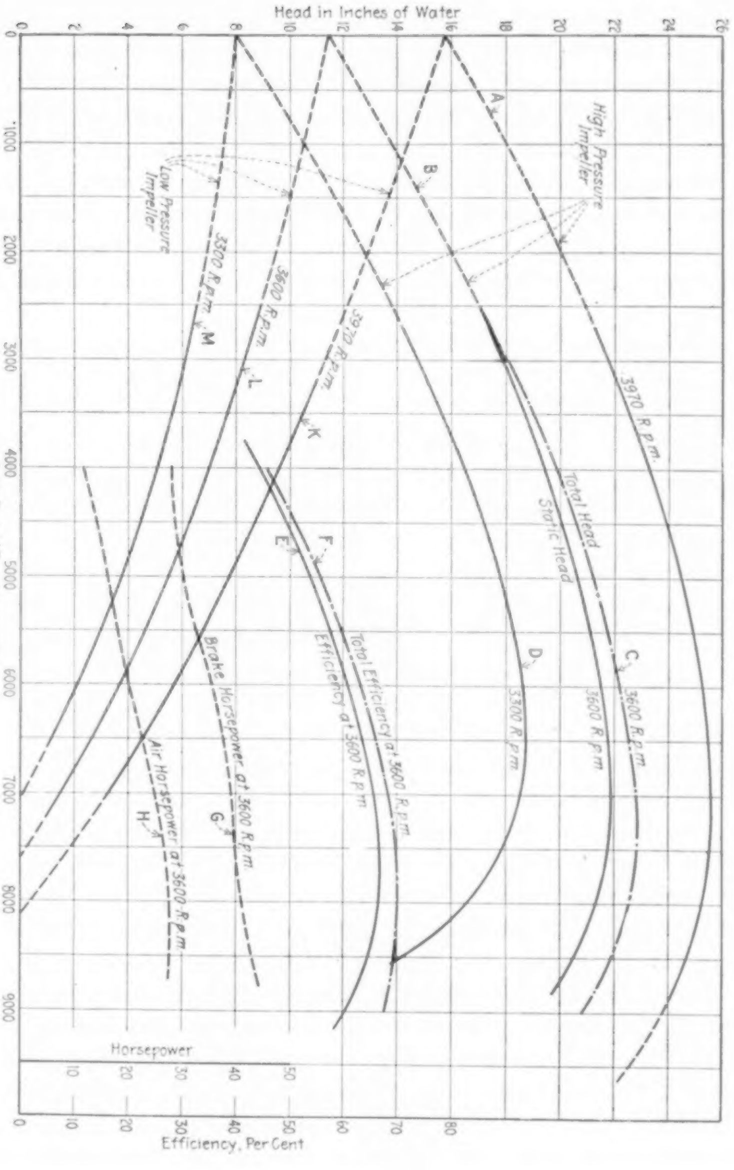

Fig. 27 Characteristics of the Two Impellers Tested

but unfortunately, to be a part of the pump, but that the predominant factor was the casing and its volute, made sometimes with and sometimes without diffuser passage.

A few years ago the writer undertook a series of experiments to show that the form of the vanes determined absolutely the characteristic curve. Some of these experiments were described in Power[1] and the test curves are reproduced in Figs. 27 and 28, and were found to match those established before the apparatus was constructed. Two fans were made to suit the same casing which was of the plain volute box form. The fans had exactly the same outside and inside diameters, viz., 18 in. and 12 in. respectively, and the width measured axially was 6 5/8 in. for both.

Thus, the runners being of the enclosed type, there was nothing outwardly to distinguish them one from the other, and as a matter of fact they differed only in the form of the blading, as shown in Fig. 28. These fans were run at 3600 r.p.m. and at that speed, for which they were designed, one gave 7000 cu. ft. of free air per minute at a pressure of 22 in. of water gage; the other, at the same speed gave 5250 cu. ft. of free air per minute at a pressure of 5 in. of water gage. At corresponding speeds it is seen that the characteristic curves of both fans began at precisely the same point, and thenceforth diverged as would of course be indicated by the proper theory.

A correctly designed impeller would work well in a casing of the box form, that is without diffuser passages, diffuser vanes or volute, provided, however, that the discharge from the periphery of the impeller is not interfered with or choked. To some extent the plain centrifugal pump of the multi-stage type is an example of this in which smooth passages are provided, but without a volute connecting one stage to the other. The efficiency of this pump is just as high as that of the double suction volute centrifugal pump.

The author gives no detail drawings of the impeller and casing and the matter as presented is vague, yet the statements of the paper are such as to amount to a claim for an important discovery, viz., that the form of casing virtually controls the performance curve of the impeller. The questions of centrifugal air compressors and pumps are of prime importance and should

[1]Horsepower of a Fan Blower, Power, June 13, 1911, p. 904.

be presented before a body like this society in a manner to permit the interested members to deal with complete fundamental data and facts which can be easily verified.

The Author. Mr. Carrier remarked about the relation of the inlet to the outlet diameter of the impeller and suggested that the efficiency of the impeller depended upon the ratio of these diameters. The author has, however, proved by experiment that this relation does not affect the efficiency of the blower materially if all other proportions of the blower are correct for the particular impeller employed.

Mr. Woodbury objects to the presentation of the author's illus-

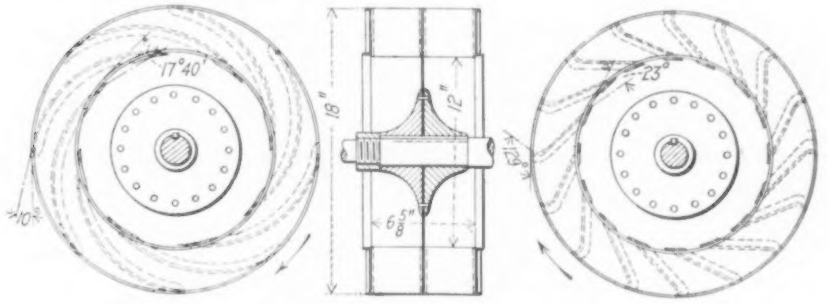

Fig. 28 Impellers Used Showing Curvature of Vanes

tration of the stream lines in an impeller: these are only diagrammatic. There will be, of course, a certain amount of compression on the driving face of the impeller blade.

In reply to Mr. R. H. Rice, it will be noted from the theory and mathematics presented in the writer's paper that his statements refer only to impellers having straight radial blades, with the exception that the blades may be curved at the inlet without materially affecting the theory as presented.

Referring to Par. 8, Mr. Rice acknowledges that without proper conversion of the velocity of the air leaving the tips of the blades into the form represented by pressure, the maximum efficiency neglecting all losses can be but 50 per cent.

Referring to Mr. Rice's discussion of Par. 11, in which he offers an analogy between the diffusion tube, or guide vanes, and the " down-stream half of a venturi meter, and we are all familiar with the fact that venturi meters are efficient for measuring

both compressible and incompressible fluids." Mr. Rice has here taken for illustration the use of the diffusion tube of a venturi meter which, as far as the author can see, has absolutely nothing to do with the use of the diffusion tubes or guide vanes in centrifugal blowers. The efficiency of the venturi tube as a meter is entirely independent (within very insignificant limits) of the efficiency of the diffusion tube in reconverting the velocity created, necessary for measurement, back into pressure. Moreover, manufacturers of venturi meters insist upon a distance of at least 20 diameters of the pipe from the nearest obstruction or elbow to the mouth of the meter, for the reason that the eddies set up by any obstruction cause fluctuations of and eddies in the flow to the meter. Because of these eddies, at least those which extend beyond the bowl of the meter, the efficiency of the diffusion tube is materially decreased, causing an unduly large drop in pressure through the meter, which is detrimental to its reputation because of the loss of head. The eddies forming above the throat cause inaccuracy in the measurements, which are also detrimental to its reputation. This is common experience in the use of venturi meters.

In Mr. Rice's criticism of Pars. 12 and 13 regarding the laws and variations of the revolutions and volume, the author wishes to say that in Par. 13 the law which Mr. Rice has stated in mathematical form, namely, $Q \div N = C$, is stated in the form, that the capacity varies almost directly as the number of revolutions per minute, which to the layman is probably more comprehensible than when expressed in mathematical form. Through an oversight in proofreading, it is stated that the work in foot-pounds per pound of air varies directly as the square of the tip speed, or if the diameter of the impellers is fixed, directly as the number of revolutions per minute. This should read, directly as the square of the number of revolutions per minute.

Further in regard to the venturi meter, Mr. Rice agrees that the diffusion tube of the meter does not affect its efficiency since he has omitted entirely any criticisms of Fig. 21, which is nothing more than a venturi meter without the diffusion tube for the simplicity of installation in a pipe line already in place.

In his conclusions regarding the efficiency of the diffusion tube of a venturi meter as applying to a blower, he disregards

the fact that the eddies in the air leaving the tips of the blades of the most perfectly designed impeller far exceed any that will be caused by an elbow or other obstruction to a venturi meter, and if it is found so important to take these precautions to avoid serious pressure drop in a venturi meter, the reader can draw his own conclusions as to how this would apply to a blower.

Regarding Mr. Rice's criticism of Fig. 6, while this figure is only diagrammatic, as has been stated, it is logically exact, although the pressure has fallen to half the maximum at the point of maximum efficiency. It is evident from the curves given in Mr. Rice's paper that in order to insure freedom from "up-setting," it is necessary to operate the blower on a comparatively small range below its normal rating and at less than maximum efficiency in order to avoid fluctuations of pressure in the discharge. The characteristics of the guide-vane blower, given by a curve such as Fig. 6, relate only to a blower of the guide-vane type, or more exactly to one in which the guide vanes have but a small clearance between their entrance edges and the impeller. However, if the guide vanes of a blower run with too small a clearance between the tips of the guide blades and the impeller, the noise which is created is of such an objectionably high pitch that it is impossible for operators either to run the machine or work at the furnaces, and for this reason it is the common practice of all builders of guide-vane machines to leave a clearance of anywhere from ½ to 2 in. between the tips of the impeller and the guide vanes, and this (as will be noted from the discussion of the free vortex in Par. 6) clearance though comparatively small radially; in high-speed blowers it forms a considerable proportion of the diameter and the theory shows that since most of the work done in a free vortex is performed in the portion near the impeller, a considerable part of the work of conversion of velocity into pressure is done before the air enters the guide vanes. As a consequence the characteristic of the guide-vane blower such as ordinarily manufactured shows less pressure rise from no-load to the point of maximum pressure, and less drop from the maximum pressure to the pressure at the point of maximum efficiency than is shown by Fig. 6.

In the discussion of diffusion tubes, the author referred only to the maximum point of efficiency, that is, at the point of break-

down for any given volume, and stated that it is possible for any given flow to have one point of steady pressure. This point, however, is always below the point of maximum efficiency.

Referring to Fig. 22 and to Fig. 10, in Par. 25, the word " the " before " characteristics " should have been omitted, as it was not intended that it should be understood that the 40,000-cu. ft. volume curve and efficiency curve referred to a guide-vane blower. The difference between the 40,000-cu. ft. and 30,000-cu. ft. blowers is simply a matter of the design of the impellers in relation to the design of the casing, the 40,000-cu. ft. blower having only a partial vortex conversion and a large percentage of velocity conversion in a diffusion tube attached to the outlet of the blower. This diffusion tube was quite efficient as sufficient conversion had been accomplished in the vortex to assure comparatively parallel stream lines at the entrance to the diffusion, but the low efficiency at small discharge rates is clearly shown by the increased pressure from no-load to the point of maximum pressure. The reason for the rapid falling off of pressure in this blower without increase of load was due to the fact that the casing of the machine was too small for the impeller.

Messrs. Moss and Crissey have also unfortunately misunderstood Par. 25 and assumed that the 40,000-cu. ft. blower was fitted with guide vanes.

Mr. Rice has stated in referring to Par. 7 that the reason for the flat pressure curve obtained on the free-vortex blower is due to the fact that there is not a proper conversion of velocity into pressure. The author made tests on this blower, obtaining the pressure at some 40 points, and has plotted a curve from the readings which shows that the pressure created in the free vortex at no-load was exactly equal to the pressure created by the impeller at no-load, and at full-load, that is, approximately 25,000 cu. ft. per min. The pressure created or the actual work done in foot-pounds per pound of air in the free vortex was about 20 per cent in excess of the work done in foot-pounds per pound of air in the impeller, which is contrary to Mr. Rice's statement. The actual efficiency of the free vortex from the tips of the blades to the discharge opening exceeding 90 per cent is a figure which has never been obtained in any diffusion tube no matter how carefully the stream lines are preserved at the entrance of the diffuser. These tests show comparatively uniform efficiency of

the free vortex from the range of no-load to the maximum capacity of the machine. The variation in efficiency from no-load to maximum efficiency does not exceed 15 per cent.

Referring to Fig. 23, showing the actual pressure in the guide-vane machine, the actual pressure curve is probably correct for properly designed guide-vane blowers, but the added line showing the pressure which would have been obtained without refinement of design, while probably true for a guide-vane blower having considerable clearance between the blower and the guide vanes, would have to have some absurd conditions in the guide vanes to show so flat a characteristic as the pressure above that due to "the pressure by perfect centrifugal action without conversion." In fact, there would have to be a great error in the angle of the guides or some obstruction or other means would have to be used to reduce the efficiency of the guide vanes in order to maintain this flat characteristic and avoid a change in the velocity conversion of the diffusion tubes as the load increases.

Returning to Mr. Rice's discussion of Fig. 6, he states that the inefficiency of the vortex type of blower is due to the loss of shaded area in Fig. 6 which gives the curve of Fig. 7. This statement, however, does not agree with the facts obtained on tests of vortex blowers of proper design, since, when the blower is tested, absolutely no variation occurs at any load except as hereafter noted.

In Fig. 24 Mr. Rice has shown a construction of radial bladed impeller designed to take in the air without shock without the use of the spiral inlet having blades extending into the hub, and the taking in of the air without shock is accomplished by means of what amounts to a propeller in the "eye" of the blower. Mr. Rice has failed to take into consideration the fact that in the construction of an impeller which extends radially inwards beyond the eye of the casing is necessarily inefficient at light-load and at overload. At light-load, because the entrance angle of the blades is designed for full-load, there is a tendency to take up or put in more air than is discharged. Consequently eddies are formed, due to the fact that the air taken into the inlet is in excess of that being discharged and, consequently by centrifugal force, is thrown out at the outer diameter of the inlet. In addition, there is a loss due to slippage, that is, the vanes pass

the air without picking it up, thus pre-heating it before entrance into the impeller proper, and consequently more work must be put into the air in foot-pound per pound than would occur if the blades did not extend into the hub. A little calculation will show that the loss due to this cause exceeds that due to the crude method employed in blowers, the blades of which do not extend to the center of the shaft or hub, for with proper design of the casing at the inlet to the blades the slippage loss at light-load is almost entirely eliminated and the loss at full and overload is for all practical purposes negligible.

In answering the discussion by Mr. de Laval in which he criticizes the use of six stages for pressures as low as 10 or 20 lb., Mr. Rice in the closure to his paper, Commercial Application of the Turbine Turbo-Compressor,[1] stated that in the later type of blowers the number of stages had been reduced from six to three and higher efficiency attained "because of the higher speed and the smaller loss resulting."

The principal advantage which Rateau, Pokorny & Wittekind and Jaeger and other manufacturers of multiple-stage machines claim is that for the higher pressures the greater the number of stages, the more perfectly the gas can be cooled, and consequently the more nearly isothermal compression is approached, with the result that higher efficiency over-all is obtained. However, Mr. Rice states that a decrease of the number of stages for the same work done is accompanied by an increase of efficiency, although the amount of cooling surface and cooling is less in a blower of a higher number of revolutions per minute. Consequently, a fewer number of stages give a better efficiency in spite of the lesser cooling, than the greater number of stages. The tip speed of the impellers must be increased as the number of stages is decreased with the result that the tip speeds necessarily are not satisfactory with the built-up construction such as shown in Mr. Rice's Fig. 24, which, though supposed to be a built-up construction, does not show the method of securing the blades on to the disc.

Further in regard to Fig. 10, it may be remarked that both Mr. Rice and Mr. Moss have credited the partially free-vortex blower with from 80 to almost 85 per cent efficiency, which might have been obtained had the turbine driving the 40,000-cu. ft.

[1] Trans. Am. Soc. M. E., vol. 33, p. 381.

blower sufficient capacity to reach the load of maximum efficiency. The reason why the 30,000-cu. ft. blower did not reach a higher efficiency than the larger was due to an oversight in the design of the rotor of the 30,000-cu. ft. blower which caused the efficiency of the impeller alone to be very low, though both the 30,000 and 40,000-cu. ft. impellers had straight radial blades without any curvature at the blade entrance, and both blowers had exactly the same casing, except for the difference in the free vortex. Had a rotor of the same design as that of the larger machine been employed in the smaller, the efficiency would have easily exceeded 80 per cent.

Since the 40,000-cu. ft. blower showed that its efficiency at its most efficient load would be 80 per cent or better, Mr. Rice's criticism of a volute and forced vortex at the inlet, combined with straight blades, is not borne out by facts.

In general Mr. Rice's remarks about a flat characteristic being due to improper velocity conversion in the free vortex are not substantiated by facts, since when the other losses in the blower are considered, an efficiency of only 74 per cent requires a very high efficiency of the free vortex and as stated, this in the 30,000-cu. ft. blower was in excess of 90 per cent. Though the velocity conversion in the 40,000-cu. ft. blower was considerably less efficient than in the 30,000-cu. ft. blower, the efficiency of the larger impeller was considerably higher, the difference in impeller efficiencies is far more than sufficient to offset the gain in the efficiency of the free vortex.

Mr. Moss states that a rising pressure characteristic is absolutely necessary for a blower of high efficiency. The author, however, can construct a free-vortex blower which will have a rising pressure characteristic, flat or rapidly dropping characteristic, using radial blades in each case, and obtain the same efficiency simply by changing the proportions of the impeller according to the characteristic desired, but as has been previously pointed out, only a blower with a nearly flat or dropping characteristic is satisfactory.

Mr. Moss says that at light loads there is enormous friction in the free vortex because "a particle must make many revolutions before it gets out." There is of course more friction per unit mass of air in the vortex at light-loads, but this loss is not so serious as Mr. Moss thinks, and at full-load in a properly

proportioned blower, the friction loss is very small indeed. Mr. Moss rightly points out that as the load increases, the path of a particle constantly becomes shorter, until at full-load it makes only part of a revolution between leaving the tip of the blades and entering the volute or collecting passage. At full-load, there is no difference between a free vortex and a perfect guide-vane blower, except the free-vortex blower does not suffer the additional friction loss caused by the surface of the vanes, since at full-load the path of the particles leaving the impeller coincides with the form of the guide vanes, or should do so, and therefore at full-load, the presence of obsence of guide vanes will have little influence on the over-all efficiency. In other words, an ideal guide-vane blower would have movable and flexible vanes which could adjust themselves to the angle of discharge from the rotor, and bend themselves into the curve of a particle in a free vortex at each particular load, in which case, again, except for the friction of the air on the surface of the vanes, the guide-vane blower would be but a less perfect free-vortex machine with a flat pressure characteristic, if properly proportioned.

One of the faults with the guide-vane blower is that only for one rate of discharge are the vanes at the proper angle; at light loads the angle is too large; and at overloads the angle is too small and likewise at only one load can the guide vanes be of the proper shape. Thus, while a straight diffuser such as the down-stream end of a venturi meter may have a high efficiency for converting kinetic into potential energy, a curved diffuser or a straight diffuser cannot be very efficient except when it conforms to the natural undisturbed stream lines of the fluid entering.

The foregoing also applies still more rigidly to the short guide vanes as used by Pokorny & Wittekind, Escher, Wyss & Company and Jaeger. Short guide vanes make a large number necessary for a given diameter and a consequently increased loss due to large friction surfaces, a greater number of edges exposed to the air stream and the breaking up of the air into numerous streams of small cross-sectional area compared with the friction surfaces encountered. In a free-vortex blower the entire mass is kept solid, which is the reason that the friction loss is so much smaller at all loads than in any form of guide-vane machine.

Messrs. Rice, Moss and Crissey have little fault to find with

the discussion of the diffusion tube and apparently accept it as correct. Mr. Moss, however, has misunderstood it. He agrees with the theory, but incorrectly interprets it by saying that "as the volume passing through the blower changes so as to change the velocity at the beginning of the tube, the blower itself produces a pressure corresponding to the new conditions." This statement would be true if the velocity of the stream of air approaching the "beginning of the tube" varied directly as the volume, or in other words if the speed of the blower always varied directly as the volume, but at a constant r.p.m. the velocity of approach varies only slightly at the tips of the blades regardless of the volume delivered, and the writer was discussing the characteristics of the two types of blowers at constant speed. Furthermore, Mr. Moss will not obtain the conditions stated, namely, constant relation of velocity of approach and pressure of equilibrium, even with a constant volume regulator, since the pressure required with the same volume will vary over wide ranges, whereas the r.p.m, varies as the square root of the work done in foot-pounds per pound of air.

The writer was misunderstood in the statement in Par. 22, "for some fixed relation between the velocity of approach and final pressure ..... constant delivery is also possible." This applies only to the points of maximum efficiency of the diffusion tube, since it is obvious that as long as the pressure in the receiver R, "which is controlled by a valve, or other means," is kept below the pressure which gives maximum efficiency of the tube for the given velocity of approach, the pressure in the receiver will remain very nearly constant.

Mr. Moss uses the same example in regard to the conversion of kinetic energy that Mr. Rice uses, namely, the action of a venturi meter, but the author has pointed out the difference between the action of a venturi meter diffuser and the conditions under which it operates, and the conditions under which the diffuser of a blower operates. As previously pointed out, the error arising from such a comparison is due to the fact that whereas in the centrifugal blower the velocity of approach before the mouth of the diffusion tube is very nearly constant, in the case of the venturi meter the velocity of approach is always directly proportional to the flow through the meter. Furthermore venturi meters as ordinarily employed handle water which

is for all practical purposes incompressible. The pressure fluctuations which might be noticeable when used for the measurement of air would not be noticeable in the handling of water.

So long as a blower fitted with guide vanes operates at a point below its maximum efficiency for any given delivery and r.p.m., pulsations of any serious nature can be entirely eliminated, as was pointed out by Mr. Rice in his paper on the Commercial Application of the Turbine Turbo-Compressor. However, since the guide-vane blower has admittedly by the discussion of Messrs. Rice, Moss and Crissey, a less efficient conversion of velocity into pressure at light-load as well as loads in excess of that for which they are designed, and that in order to avoid fluctuations of pressure, the efficiency must be still further reduced, is sufficient evidence in itself to prove that a free-vortex blower, which is capable of attaining an efficiency of 75 to 80 per cent at full-load with rising pressure characteristic from full to no-load, is far more desirable than a blower fitted with guide vanes, which under any conditions whatsoever necessitates the use of a speed regulating mechanism in order to give it the slightly decreasing pressure characteristic obtained in a free-vortex blower at constant speed. In other words, where a very rapidly decreasing pressure characteristic is essential, it is evident that since the free-vortex blower has of itself a decreasing pressure characteristic at constant speed a smaller variation of speed is necessary in order to give any given slope of pressure characteristic than is required in the guide-vane blower. This means of course a higher efficiency of the driving turbine or motor.

Mr. Moss next takes up the mathematical part of the paper, showing how this can be very much simplified. The several reasons which he presents are perfectly correct. In such a deduction as Mr. Moss has given, for instance, of the centrifugal compression in the impeller, it would be difficult for the reader to understand the explanation unless he was already familiar with exactly what takes place in the impeller of a blower. The same applies to the deduction of what happens in a free vortex as it would be very simple to show that, since the velocity in the free vortex is inversely proportional to the radius it is evident for instance that if a point is taken at twice the radius of the impeller, the velocity would be one-half that at the blade tips, and

therefore, its energy one-fourth, whence three-quarters of the energy at the tips of the blades would have been converted into the form of potential energy as represented by an increase of pressure. The same also applies to the analysis of the diffusion tube for Bernouilli's theorem states that the velocity varies inversely as the area and consequently for an incompressible fluid (or small pressure variation in a compressible fluid), it is evident that if the mouth, or large end of the diffusion tube is twice the area of the inlet end, the velocity must be one-half that at the inlet, and consequently has kinetic energy only one-fourth as great as at the inlet. Since only one-fourth of the kinetic energy is left at the mouth of the diffusion, three-fourths of the kinetic energy must be converted theoretically at least into the potential form, namely, by a rise of pressure. The author would call Mr. Moss's attention to the fact that in the equation

$$Q = 6540 \frac{A\,T_2}{\sqrt{T_1}} \cdot \sqrt{(K+1)\,K}$$

the constant is not $60\sqrt{2\,g\,J\,C}$ since the mechanical equivalent of heat J does not appear in this formula. Also, in Mr. Moss's equation for the work done by adiabatic expansion, the mechanical equivalent of heat should be omitted.

Mr. Moss also gives a formula for the horsepower in which he uses the author's K the absolute inlet pressure and a constant, $\mathrm{H} = \frac{P_2 K}{3330} \frac{\gamma}{\gamma - 1}$. The constant 3330 in this formula appears to be incorrect and since γ is essentially constant for all the so-called perfect gases, the author does not see why Mr. Moss did not include $\frac{\gamma}{\gamma - 1}$ in his constant 3330. Mr. Moss's criticism of the author's long equations is not justified inasmuch as the layman in order to see exactly the processes which are going on during the operations in a blower, must see these in the differential form. For those familiar with the actions in a blower, the shortened processes, of course, are equally clear. Had this part of the paper been published in its entirety, the criticism of the long formula would not have been necessary, as the author, in the main part of the text, even omitted the cancellation of similar terms of the numerator and denominator of the various equations in order that the reader could follow through in easy steps the derivation of the final equation.

Because of this criticism the author here presents the contracted formula relating to the adiabatic expansion and compression of air and curves of the value of $\left[\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}-1\right]=K$ on a scale sufficiently large that the value of K and the value of $\frac{P_1}{P_2}$ can be found with sufficient accuracy for all practical and experimental purposes, and can likewise be used for solving the equation for the pressure correction given in Par. 68. It will be necessary in this case to work backward from K to the value of $\frac{P_1}{P_2}$ on the diagrams, imagining a 1 in front of the decimal point of the values of K.

Mr. Crissey has criticized the author and stated that he has evidently compared the free-vortex blower with turbo-blowers of inferior design, and that sharp peak and rapid drop of efficiency on either side of the peak are apparently characteristic of poor design. However, the writer has failed to find any authentic tests of guide-vane blowers which show a smoother efficiency characteristic than that shown, in spite of the fact that some of the tests of guide-vane blowers which he has examined indicate an efficiency as high as 78 per cent.

Mr. Crissey states that " every builder of turbo-blowers, turbo-pumps and turbo-compressors would like to omit diffusion vanes because they add to the expense of production, but experience has shown that they are advantageous. Their use led to the introduction of high-pressure centrifugal pumps, etc." He will find, however, that there are other reasons for the use of the vanes, namely, the introduction of guide vanes as an easy means of overcoming difficulties which were at that time not really understood by designers, but perfectly well known to those sufficiently informed on the theory of centrifugal compressing and pumping apparatus.

Referring to Mr. Crissey's discussion of the impeller construction, he together with Messrs. Rice and Moss, apparently entirely misunderstood the author's meaning in regard to straight radial vanes and what was meant by " the lowest blade speeds."

For tip speeds less than 300 or 400 ft. per sec. built-up construction may be satisfactory provided the blades are not too

broad in axial direction so that the unsupported surface in proportion to the work done by one blade is not too great. Where the blade speed exceeds 300 or 400 ft. per sec., built-up construction may be satisfactory provided the blades are very narrow in an axial direction and that the work done per blade is small. Mr. Rice has referred to the fact that of the hundreds of machines which have been built in Europe only a few are in operation, and has suggested that this is due to the method of balancing employed by Rateau, but as a matter of fact Mr. Rice will find that most of the blowers which are not operated are out of commission because of improper impeller construction.

Mr. Crissey remarked that manufacturers who have employed milled-out construction have abandoned it and returned to the built-up construction, but he has omitted to state that this was done because the built-up construction is much cheaper than the solid construction. The author is familiar with a number of cases where impellers running at only a moderate speed of between 300 and 500 ft. per sec. have vibrated to such an extent as to loosen the riveting and wreck the machines. In speeds exceeding 500 ft. per sec. and any considerable blade width, except for the commercial reason of cheapness of manufacture, there is no logical excuse for using anything but the solid rotor construction.

Mr. Crissey states that he prefers the European method of testing to author's; that is, the measurement of the air at the intake of the blower instead of at the discharge, and gives as one of his objections, that it necessitates correcting the volume of air from the discharge pressure and temperature at the mouth of the nozzle to that of the atmospheric pressure and temperature. To one designing blowers so simple a problem should not present such great difficulty and so great a source of probable error.

Following is a standard clause of all blower contracts of The Westinghouse Machine Company, showing the method of conducting tests, and Mr. Crissey will note that the volume delivered by the blower is not only measured at the intake but at the discharge as well, and the volume as measured at the discharge must check within one-half of 1 per cent of the volume as measured at the intake. The writer believes this eliminates any chance of error.

METHOD OF TESTING HIGH PRESSURE CENTRIFUGAL BLOWERS

The method of test which shall constitute part of this contract will be the operation of the blower at constant speed. The volume of air handled and the efficiency of the blower will be calculated from the equations given on Drawing 64048 attached; the readings of pressure and temperature to be taken as shown in Fig. 29.

As shown in Fig. 29, the volume will be checked by three methods, viz.: the flow as calculated through a nozzle on the intake to the blower, flow as calculated from a nozzle on the discharge of the blower located approximately as shown in the drawing, and as a check against the volumes as measured from the intake and discharge nozzles, the volume will likewise be checked by the difference between the static pressure measured at the discharge, and the total (or static plus velocity pressure) as measured by a Pitot tube, as shown at the upper part of Fig. 29. The volume actually handled shall be considered as equal to the mean between the volume measured at the inlet nozzle and that measured at the discharge when the latter has been corrected to the inlet conditions, according to the formula given on Ddawing 64048. The pressure at the throat, or minimum section of the intake nozzle, shall be taken at four points as shown attached hereto.

In the intake a Pitot tube shall be inserted, as shown in Fig. 30, and readings taken at each 1 in. of the radius, the first reading being taken ½ in. from the side of the throat. The temperature shall be taken at four points at arbitrary radii, as indicated by the thermometer on Fig. 29.

The mean static pressure at the discharge in the blower and the mean temperature shall be taken by means of four thermometers and four pressure connections, arranged similarly to Fig. 50, except that Pitot tube readings need not necessarily be taken at this point.

The co-efficient of flow for the intake shall be taken as 0.995, and the co-efficient for the discharge shall be taken as 0.99.

If, without resorting to the Pitot tube readings, the volumes, when reduced to a common basis as calculated from the intake and discharge orifice, do not check within one-half of one per cent, both the intake and the discharge shall be searched by means of a Pitot tube, as indicated in Fig. 30, and other discharge co-efficients calculated from the Pitot tube readings.

As the intake nozzle and the discharge nozzle are of very considerably different diameters, if, assuming the above co-efficients, the flow reduced to a common basis checks within one-half of one per cent, this would in itself constitute a proof of the correctness of the method of measurement, since, if there is any considerable correction for the coefficient of contraction of flow through an orifice when constructed in accordance with the drawings attached, the co-efficient of contraction would vary considerably with the diameter, and consequently the flow, as calculated from two nozzles of different diameters, could not possibly check within the limits above described.

If the barometric pressure existing during the trials, and the temperature of the air at the inlet varies from that specified in the contract, the

correction of the discharge pressure for the barometric reading and the inlet temperature shall be made in accordance with the formula P'.

$$P'_1 = P'_2 \left\{ \frac{T_2}{T'_2} \left[\left(\frac{P_1}{P_2} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] + 1 \right\}^{\frac{\gamma}{\gamma-1}}$$

this formula for correcting the discharge pressure being used only should

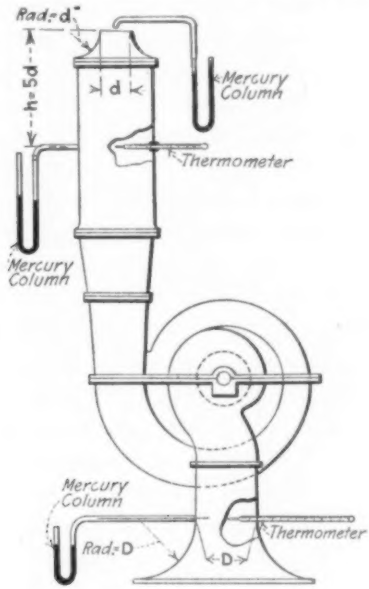

FIG. 29 DIAGRAMMATIC ARRANGEMENT OF CENTRIFUGAL BLOWER TESTING APPARATUS

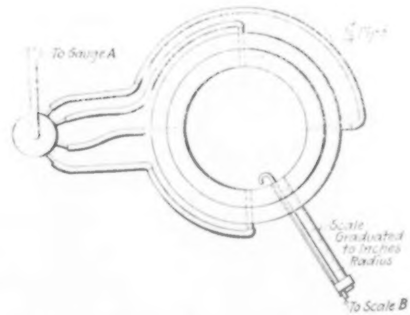

FIG. 30 PLAN VIEW OF BLOWER INLET

the pressure on test fall below that called for in the contract, the efficiency being independent of the temperature at the inlet.

In any blower operated continuously at no-load, whether of the guide-vane or of the free-vortex type, the temperature of the air rises owing to friction losses in the blower, and as the temperature rises the density of the air decreases. Consequently, since for a given tip speed or constant r.p.m., only a fixed number of foot-pounds of work per pound of air can be done in the impeller, there will be a rise and fall of pressure due to the fact that as the temperature increases, part of the air within the casing will back into the intake, and will be replaced by cooler air by mixing. This will again cause the pressure to increase owing to the greater density of the cooler air, and the blower to rise to a higher pressure than it can maintain; but it is only at no delivery that this phenomenon occurs in a free-vortex blower, whereas, unless run at an overload for a given r.p.m., this occurs at all loads on a guide-vane blower.

Mr. Crissey gives a formula for equivalent weight of air in pounds per second entering the blower inlet, but the author believes that he will find that this is rather more complicated than the contracted formula when used with the curves belonging to it. Apparently from the nomenclature employed by Mr. Crissey, it is evident that he follows the directions furnished with the license to build Pokorny & Wittikind's blowers.

The author wishes to thank Mr. Crissey for calling his attention to the errors in Par. 13, and in Fig. 10, as both the pressure and efficiency curve of the 30,000-cu. ft. blower should have been lettered 3600 r.p.m.

In regard to the symbol P in general throughout the paper, this refers to the static pressure, but it may also be used under some circumstances as a static plus the velocity head. P_2 is always the inlet pressure to the blower, or the inlet pressure of the impeller of any given stage of a multiple-stage blower under consideration, and T_2 is always the absolute temperature at the entrance to the inlet of the blower, or the inlet of any given stage of the multiple-stage blower under consideration.

Mr. de Laval refers to some works on blowers with which the author is familiar, but their discussion is impossible at this time owing to the fact that there are so many discrepancies, and in some cases, falsely assumed conditions employed in the various articles and books referred to, that space would not permit even of pointing these out.

Mr. de Laval states " that the experiments by Rateau, Parsons, etc., distinctly point to the rational design of diffusers as solving this problem." He refers to Parson's experiments and says that after building Parson blowers of the axial compression type he found that the centrifugal blower was more efficient and proceeded to produce similar blowers. The axial compression blower of Parson's depends entirely upon the velocity conversion in curved diffusion tubes, namely, the stationary vanes located between the moving vanes of the blower, and also in the moving blades. As the low efficiency of the Parson blower is well known, Mr. de Laval has selected a very poor reference in proving the efficiency of guide vanes.

He likewise says that the calculation of a centrifugal compressor is similar to that of a centrifugal pump and gives the formula $p = \frac{v^2 s}{2g}$. This same error occurs in the book of " Ostertag," to which Mr. Crissey referred and is absolutely incorrect, for anything but a very low pressure.

Mr. de Laval has evidently entirely overlooked the true meaning of the deduction of the author's equation

$$\frac{T_1 - T_2}{T_1' - T_2}$$

inasmuch as the derivation of this formula was simply to show that regardless of what method, or from what source the losses occurred, it still holds true. It was only to state this in differential form so that the processes at every point and instant could be appreciated by the reader that this deduction was made. It is evident that if the law of conservation of energy is granted, it is necessary that the heat put in at the shaft must appear as a sum of the heat reappearing as useful work plus the heat wasted in other ways. This in itself is a proof, but is not so easily understood.

Mr. de Laval states that this method of testing is not correct because of the fact that there is some radiation from the casing of the blower, and also some heat lost in the bearings which does not appear at the discharge of the blower. This is correct, but he will find that compared with the total heat energy going through the blower in a given time the small area exposed for radiation and loss of heat by convection and the power lost in the bearings is so small that it is within the limits of the errors

of observation in ordinary commercial tests. This is especially true on blowers designed like those of the Westinghouse Machine Company where the greater part of all the bearing friction heat equivalent is conducted by the metal to the intake, which warms the air, and where part of the radiation is eliminated due to the fact that the intake surrounds a large proportion of the casing of the blower. For these reasons, for all practical purposes this method, if carefully employed, is probably more accurate than measuring the horsepower and volume, pressure and calculating the efficiency from these quantities.

One serious objection, however, to the temperature method of determining the efficiency is that the greatest care must be taken to insure that warm air, as for instance from under the engine-room floor, does not enter the intake of the blower unless the blower is so constructed that it is possible to get the average temperature of the air after entering the inlet. This is the reason for the difference between the efficiency as measured from the b.h.p. input and the temperature method in the example of the blower tests given in the paper. The low efficiency shown by the temperature method was due to the fact that warm air in considerable quantities was drawn in from under the power house, whereas the inlet temperature was taken in the passage leading to the blower and the readings taken at this point were considerably lower than the average actual temperature of the air entering the blower. It was rather interesting in this particular instance to note that whether the temperature method showed a higher efficiency than the b.h.p. method, or vice versa, depended upon the direction of the wind prevalent at the time the test was being run, as this varied the amount of warm air drawn in from the power house basement.

Mr. Guy has entered upon a discussion of the relation of the impeller and casing. He puts forth the theory that the entire action of the air or fluid in a blower or compressor depends practically only upon the design of the impeller. Mr. Guy lays particular stress upon the shaping of the blade between the inlet and outlet, but the author believes that provided other dimensions of the rotor are properly made, the actual form of the curve between the inlet and outlet has very little influence upon the efficiency of the blower.

The author agrees with Mr. Guy in that any formula which

involves only the inlet and outlet angles of an impeller without other considerations will not give satisfactory results. Of all the works he has ever reviewed on the subject of the design of pumps or blowers with curved vanes, he has not found one formula which agrees with practice. As Mr. Guy says, the area of the passages through the impeller between the inlet and outlet determines its characteristics and its efficiency. That is the efficiency of the impeller alone.

However, Mr. Guy is mistaken as to the importance of the design of the impeller, since with a suitable design of casing and proper proportions of impeller it is possible to use vanes of any reasonable shape, either tipped backwards radially or forwards, and still obtain the same efficiency, provided that the proportions and design of the casing are suitable for the particular type of impeller employed. Mr. Guy is also in error in regard to the effect which the casing or its form has upon the performance of a blower, regardless of the type of impeller employed. The author has tests on low-pressure blowers which show that the characteristics of the impeller itself can be totally changed by a very slight alteration in the form of the casing. Likewise, the capacity at which the maximum efficiency occurs and the maximum capacity which can be obtained from the blower with a free discharge is determined by the casing more than by the impeller itself.

It may be interesting to note in this connection that the writer has run a very large number of tests on low-pressure blowers in which the actual static pressure at the tip of the blades was less than that at the inlet and the air entering the impeller or rotor revolved in the opposite direction to the rotation of the impeller. The explanation of this phenomenon, which so far as the author knows, has never been observed in any previous tests on centrifugal blowers, is due to the fact that the pressure head created by the impeller itself was not sufficient to balance the kinetic energy required to produce the velocity of entrance into the impeller, the difference between the pressure head created in the impeller and that necessary to cause the entrance into the impeller being produced entirely by the casing. In spite of this fact, an efficiency of over 65 per cent was obtained under this condition. The efficiency of the impeller itself being a minus quantity and the entire useful work being accomplished in free vortex. This

should also be sufficient proof to convince Messrs. Rice, Moss and Crissey of the efficiency of a free vortex when properly designed, and furthermore, this rotor had a decreasing or drooping pressure characteristic.

In regard to Mr. Guy's remark concerning "the blower described is the same as that which forms the subject of the discussion of R. N. Ehrhart on the paper on turbo-compressers[1] is simply an enlarged fan of the type as described and made by Professor Rateau some years ago," the author wishes to call his attention to the fact that the fan made by Professor Rateau some years ago was totally different from that of which Mr. Ehrhart spoke.

At this point it might be well to point out to Mr. Guy that in the opening paragraph of the author's paper, he stated he claimed no originality whatsoever for the matter presented. Mr. de Laval's statement that the influence of the shape of the casing was pointed out by Poillon in a book on Pumping Machinery about 30 years ago, is rather a recent reference as this matter was fully investigated by the late Lord Kelvin at a much earlier date and has been on record, but apparently there have been few engineers who have sufficiently understood Lord Kelvin's deductions to put them into practise.

Regarding Mr. Guy's Figs. 27 and 28 and his reference to the fact that two impellers with differently shaped vanes produced the same pressure at no-load was very interesting. The impeller with vanes curved backwards and that with the vanes curved forwards show distinctly that the box form of casing employed by Mr. Guy is extremely inefficient at light-load. With the vanes tipped backwards it is true that within reasonable limits, if the outlet is not choked in any manner by making the outlet angle sufficiently small and running at a high r.p.m. compared with the work done in order to reduce the disc friction or rotation losses, it is possible to get a very high efficiency without any means whatsoever of reconverting the final velocity into pressure. In fact, if it were not for friction, if the vanes made a zero angle with the tangent at the periphery, 100 per cent efficiency would be theoretically possible without any casing or other means of reconverting velocity into pressure. The speed for any pressure would however, have to be infinity in this case.

[1] Trans. Am. Soc. M. E. vol. 33, p. 396

Unfortunately, however, the tip speed required even in a radial-blade impeller to create any considerable air pressure is so high that curved vanes which make a small angle with the periphery are entirely out of the question for practical work. Mr. Guy has not given the efficiency of the impeller with the vanes tipped backwards. His b. h. p. and efficiency curves for the high-pressure impeller, e.g., the one with blades tipped forward, do not agree as there is a hump in the horsepower curve, whereas the efficiency and pressure curves are shown to be smooth. Some great phenomenon must have occurred at this point in order to produce such results, but on the face of it there does not seem to be any ground for thinking so.

In spite of the fact that at no-load there is theoretically no circulation of air through the impellers, one would expect that with the vanes tipped forward, due to the eddies which occur, a higher pressure would be obtained than with the blades turned backwards, though Mr. Guy has drawn his curves very nicely to agree with the theoretical case of no delivery when including the air which is revolving inside the impeller due to friction, the head created by any impeller regardless of the shape or form of the blades is exactly the same.

THE LUBRICATING VALUE OF CUP GREASES

By A. L. Westcott,[1] Columbia, Mo.

Non-Member

ABSTRACT OF PAPER

Cup greases are made by a process of saponifying animal or vegetable oil, as cotton seed, lard or tallow. To the soap thus formed is added mineral oil, sufficient to give the desired consistency. Manufacturers supply greases of several consistencies adapted to different conditions of lubrication, each being designated by a number.

2 While grease lubrication has a wide general application, it also has its own special field where it has a marked advantage over oil. This field is machinery in which the operation is intermittent, such as cranes and hoisting machinery. Grease cups of proper design will supply ample lubrication to the bearings of such a machine. They require no attention except to be filled with grease when empty and, if of the hand operated compression type, grease has to be forced into the bearing perhaps once or twice a day. No lubricant is wasted when the machine is not in operation, as is likely to be the case if sight feed oil cups and liquid lubricant are used.

3 It was the purpose of the investigations which form the basis of this paper to test a number of greases under a variety of conditions as to bearing pressure, temperature, and method of application, for coefficient of friction and general suitability as a lubricant.

DESCRIPTION OF TESTING MACHINE

4 Tests for the coefficient of friction were made upon a Golden oil testing machine. Fig. 1 is a general view of this machine, and the side and end elevations are shown in Fig. 2. The bearing consists of a babbitted sleeve which is fitted to a shaft *H*, Fig.

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Presented at the St. Louis Meeting, February 5, 1913, of The American Society of Mechanical Engineers. The complete paper may be consulted at the rooms of the Society, 29 West 39th Street, New York.

2. This shaft runs in roller bearings F and is driven by a motor through the coupling L. The motor is arranged with a reversing switch so that it may be run in either direction as desired. A cast-iron beam A is bored out at the center of its length to fit the bearing, to which it is fastened by screws. The ends of this beam are circular arcs struck from the shaft center. Flexible bands C are attached to A and support the weights B by means of

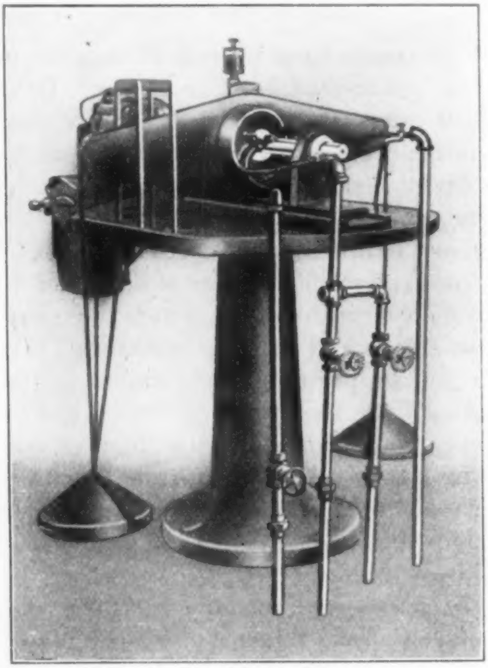


FIG. 1 GOLDEN OIL TESTING MACHINE

which the load on the bearing is applied. The casting which forms the bearing is cored out so as to provide a space for a jacket in which cold or hot water or steam may be circulated in order to control the temperature. At D a spring balance is supported upon four vertical rods that are screwed into the machine top. This balance is connected by a thin strip of spring steel to the post with screw adjustment at G. The steel strip leads over a part M, cast on the upper side of A, and machined to form a circular arc, the center of which is the center of the shaft. The radius of this arc is 6 in. and the height of the balance is such that it exerts a pull always in a horizontal direction. The weight

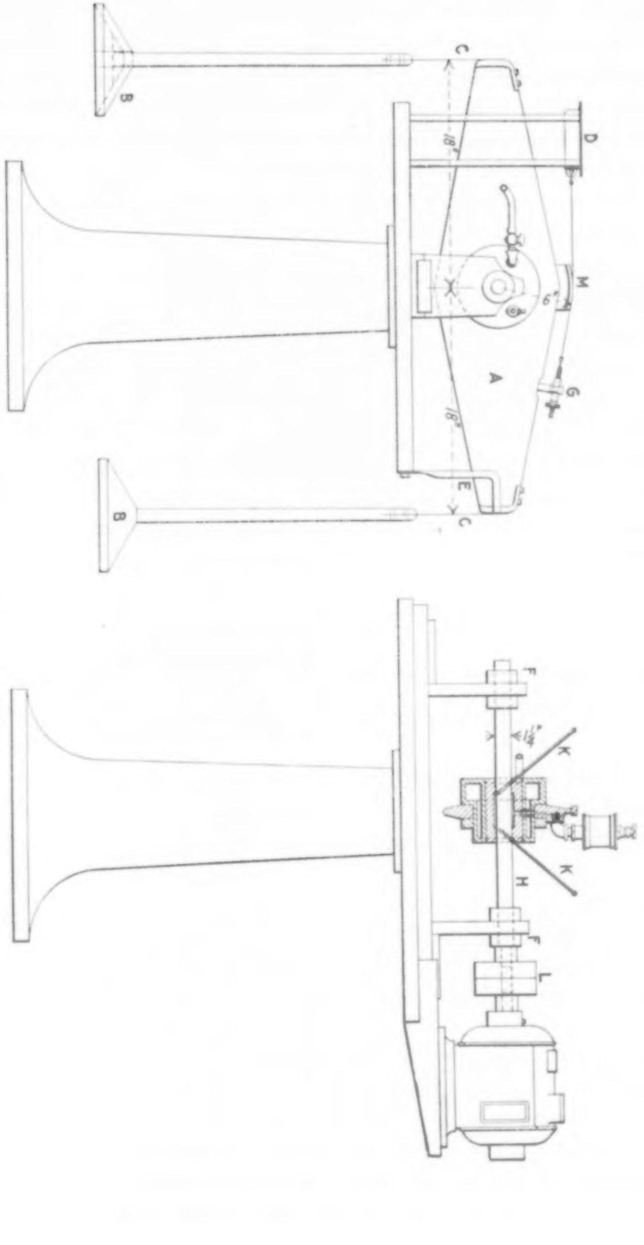


Fig. 2 Golden Oil Testing Machine

on the end of the beam opposite to D must be made a little greater than that on the adjacent end, so that a positive pull will always be exerted in whichever direction the motor is running. A pointer E, attached to one end of the table top, enables the operator to bring the beam to any desired angular position. Holes are drilled in each end of the bearing for the insertion of thermometers K, the bulbs of which are brought close to the side of the journals, so as to obtain an indication of the bearing temperature as nearly correct as possible.

5 As originally constructed, the machine differed in several details from that shown in Fig. 2.

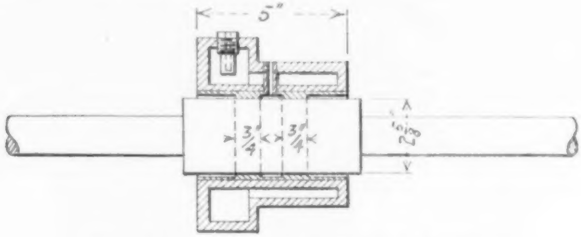


FIG. 3 , BEARING FOR GREASE TESTING

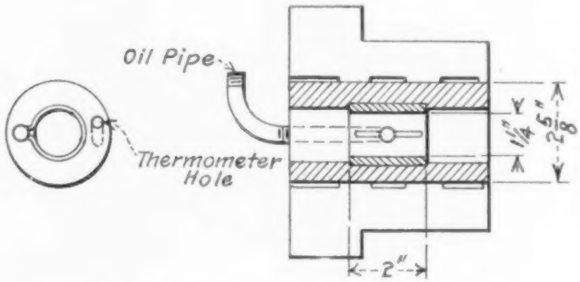


FIG. 4 BEARING FOR GREASE TESTING

6 The bearing as supplied by the makers was 2⅝ in. in diameter, by 5 in. long. With the largest load that the suspending chains would safely carry this gave only about 50 lb. per sq. in. of projected area. It was desired to carry the bearing pressures much higher than this. The length of bearing was therefore reduced to 1½ in. as shown in Fig. 3, there being two rings of ¾ in. in length each. The temperature was determined by a thermometer inserted into the jacket as shown in Fig. 3, which also shows the manner of applying the lubricant.

7 The bearing of Fig. 3 was not satisfactory. It appeared

that the proportion of length to diameter was bad, at least for grease lubrication. The grease was applied by means of a hand operated grease cup of the compression type. The counterbore between the two bearings, where the grease was forced in, was large enough to form a reservoir, and it was thought that the grease would feed in so as to maintain constant lubricating conditions. This did not prove to be the case. After an application of grease the friction, momentarily reduced thereby, would begin to increase steadily. Under these conditions a plot of friction against time would look like the profile of a rip saw; a series of lines inclined more or less steeply to the horizontal, and connected by verticals where the grease cup was operated.

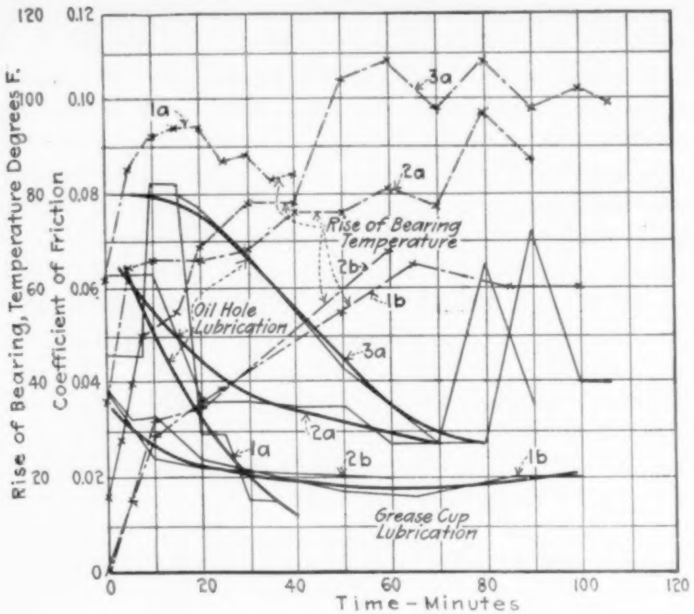
8 With a view to obtaining a more satisfactory form of bearing, the change indicated by Fig. 4 was made. The diameter was reduced to $1\frac{1}{4}$ in. and the length increased to 2 in. A sleeve made to fit the old bearing was bored out and babbitted to this size. The lubricant was applied at one side through a hole running lengthwise of the sleeve and was distributed along the journal by a groove, as shown in the figure. A hole was drilled for the insertion of a thermometer close to the journal. The temperature was controlled by the same water jacket as was used at first (Table 2).

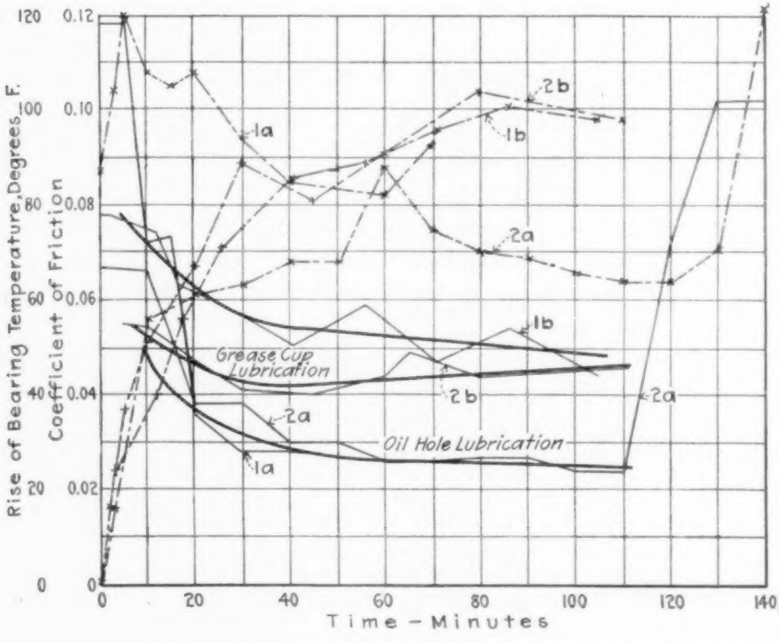
CLASSIFICATION OF TESTS

9 Three series of grease tests are included in this paper and in addition a series of oil tests was run. The grease tests will be classified as follows:

- *A* Tests to determine the coefficient of friction of cup greases of different densities, using hand operated grease cup, with intermittent feed, the bearing of Fig. 3 being employed.
- *B* Tests with the same objects as in series *A*, but with the bearing of Fig. 4 employed, and for the most part using a grease cup having a constant feed.
- *C* A series of tests with six different greases to determine their behavior when applied in a grease cellar on top of the journal, with a view to their flowing into the bearing by gravity when warmed.

10 Greases from two makers designated as *X* and *Y* were tested in series *A* and *B*; greases from six different makers, designated by numbers 1 to 6, were tested in series *C*.





FIGS. 5 AND 6 GREASE CUP AND OIL LUBRICATION COMPARED

11 The consistency of the greases designated by X is indicated by numbers: No. 00 being the hardest; No. 6 a semi-liquid, and the other numbers forming a graded series between. Greases designated by Y are likewise numbered, but their hardest grease has the highest number, while No. 1 is the softest, almost a semi-liquid, at ordinary temperatures; No. 5 Y is of about the same consistency as No. 00 X.

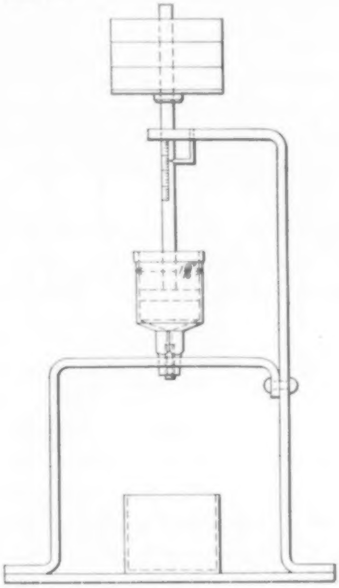


FIG. 7 APPARATUS FOR TESTING VISCOSITY OF GREASES

12 The greases of series C may be described as follows:

No. 1 soft spongy fibrous in appearance, bright yellow color
No. 2 hard bright yellow in color
No. 3 soft, dark brown in color
No. 4 very hard, light brown color
No. 5 extremely hard, looks and smells like soap
No. 6 a graphite grease, soft, black

DESCRIPTION OF TESTS

13 The tests upon each grease in series A and B were carried out as follows: The load on the bearing was varied for each grease through a range of 46 to 148 lb. per sq. in. in series A, and 53 to 154 lb. in series B. For each load, the test was made at temperatures varying through a wide range; beginning the test

at a low temperature, which was gradually increased by successive increments as the test proceeded. During the first three tests of series *A*, steam was not available for use in the bearing jacket, the elevation of temperature in these cases being due to the heat generated by friction. In series *B*, and in other tests of series *A*, however, the temperature was controlled by either water or steam in the jacket. The tests show in each

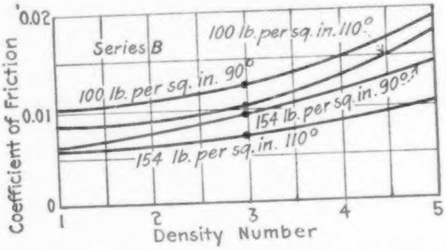


FIG. 8 FINAL SET OF DERIVED DATA FOR *B* GREASE

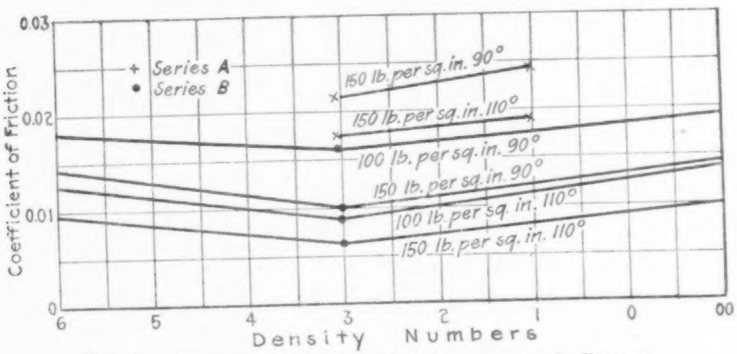


FIG. 9 FINAL SET OF DERIVED DATA FOR *A* GREASE

case the effect of rising temperature upon the coefficient of friction.

14 The results of the tests of one grease of series *A* are presented in Table 3, an ordinary engine oil, Table 4, being taken for the purpose of comparing it with grease as a lubricant under the same conditions. The tests of one grease of series *B* are found in Table 4 and Figs. 12 and 13.

15 For series *A* and *B* the plot was made of ϕ vs. bearing temperature. From these curves were derived the ordinates necessary for the next set of curves, ϕ vs. load, pounds per square inch on the bearing. Each of these was plotted for a constant temperature, curves for two or more temperatures being employed.

TEST DATA

16 All the tests of series C were made at one bearing pressure, 114 lb. per sq. in. of projected area of journal. These tests were run upon the bearing shown in Fig. 3. Two oil holes ¼ in. in diameter were drilled through the bearing sleeve at angles of 30 deg. above the horizontal and on opposite sides of the vertical center line. To perform a test a measured quantity of grease was placed in the holes, the bearing was loaded to the desired pressure, and the test was started and continued without applying any additional lubricant until the lubrication failed. Neither water nor steam was circulated through the jacket space, and the rise of temperature, therefore, was due solely to heat generated by journal friction.

17 For the purpose of comparing this method of applying the grease with forced lubrication, tests were made of the same grease at the same bearing pressures, but feeding the grease by means of a compression grease cup.

18 The curves of this series were plotted differently from those of A and B (Cp. Figs. 8 to 11). Since the length of time a given test was continued without renewal of the grease in the oil holes is an important factor in determining the value of each grease, time was plotted as abscissae, and ϕ and rise of bearing temperature as ordinates. Each curve set shows the complete test of a grease, comprising one, two or three runs made on different days.

19 Figs. 8 and 9 show the final set of derived data as to series A and B. For a selected bearing pressure and temperature, the values of ϕ for each grease were plotted against the grease numbers as abscissae, the idea being to show the most advantageous grease consistency for a given condition. Plots were thus made for two pressures, and for two temperatures for each pressure.

20 An inspection of the tables and curves brings out some well defined relations. The values of the coefficient of friction shown in series A, made upon a bearing of large diameter and small length, are greatly in excess of values for the same grease when tested on the bearing shown in Fig. 4, series B; for example, compare Tables 2 and 3, at same loads and temperatures. As has been stated before, the large and short bearing consisting of two lengths of ¾ in. each proved to be a poor form for grease lubrication. The film of lubricant seemed to have very little endurance, and the only way to get results, particularly with the

denser greases, was to force in grease frequently. Another difference in conditions between the tests of series *A* and *B* is the difference in velocity of journal. In series *A* the surface speed of journal was about double that of series *B*. The relations between speed of journal and coefficient of friction are discussed in Par. 26.

21 The tests of series *C* indicate that generally the oil hole method of applying grease to the bearing is inferior to the method of forcing it in by means of a grease cup, the coefficients of friction in the former case being much larger than in the latter. An exception is noted in case of grease No. 4, where the advantage lies with the oil hole method. If instead of a small oil hole, a grease reservoir extending the length of the journal had been used, it is possible that the results of the comparison might be different.

22 The best results so far as producing constant and uniform conditions of lubrication are concerned, were obtained when using a grease cup with a plunger actuated by a helical spring so as to give a constant feed. When adjusted to feed steadily and uniformly, the coefficient of friction at a given load and temperature remained about constant. With the intermittent feed of the hand operated grease cup, results in this regard were not so good. In many of the tests where observations were taken immediately after feeding the grease and at 1-minute intervals thereafter, the friction was seen steadily to increase even with the bearing of the form of Fig. 4, and to decrease again to its former value upon again forcing in grease. This is illustrated by the abstracts from the original log of tests of *B* greases No. 5, given in Table 1. It will be noticed that the variation in friction referred to is much more marked at low temperatures than at high ones, presumably because the grease flows better when hot.

23 The tests of engine oil, Table 3, which are inserted for the purpose of comparison of oil with grease, were run on the bearing shown in Fig. 3 (Cp. also Figs. 5 and 6).

24 The formula for coefficient of friction is deduced as follows. Let

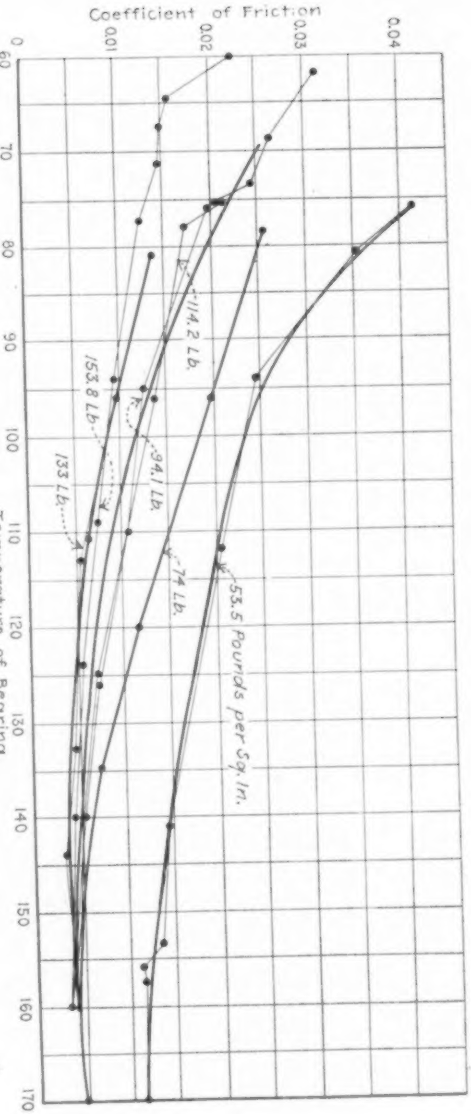
W = total load on the bearing, pounds

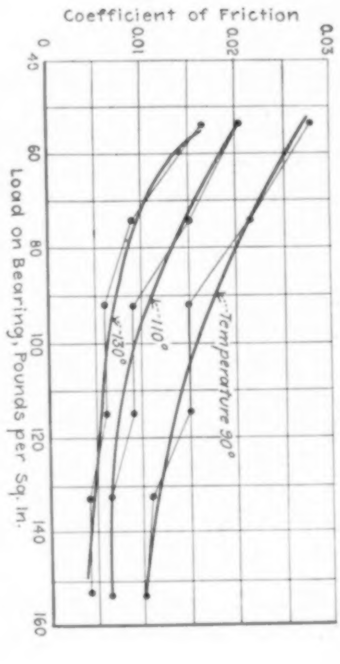
P_1 = pull on the spring balance when running in a clockwise direction, pounds

P_2 = pull on balance when running anti-clockwise, pounds

D = diameter of journal, inches

ϕ = coefficient of friction





FIGS. 12 AND 13 SERIES *B* TESTS, *X* GREASE

$$\phi\, W \frac{D}{2} = \frac{P_1 - P_2}{2}\, 6$$

$$\phi = (P_1 - P_2)\, \frac{6}{W\, D}$$

For any load W, the constant $\frac{6}{W\, D}$ may be computed once for all, and the formula takes the form

$$\phi = K\, (P_1 - P_2)$$

25 The effect of a rise of temperature upon the coefficient of friction is almost always the same. The curves show this very clearly. The fluidity of a lubricant is increased by warming it, and its viscosity is decreased. This results in decreased friction up to that point where the bearing pressure is sufficient to overcome the tenacity of the oil film, so that there is contact between the rubbing surfaces. The tests do not indicate that, within the limits of bearing pressure which obtained, there is any disadvantage or danger of cutting the bearing incident to a temperature of 150 deg. fahr. In some instances the temperature was carried as high as 200 deg. and as long as there was an ample supply of lubricant to the bearing, no harmful effects were noted. The "hot box" of practice occurs because the lubrication of the bearing has failed; the former being the effect of increased friction due to the latter. There is nothing intrinsically objectionable in a bearing temperature much higher than is commonly permitted in practice, if good lubrication obtains; and these experiments show that much may be gained in the way of decreasing the lost work of friction.

SPEED OF JOURNAL AND COEFFICIENT OF FRICTION

26 An effort was made to determine the relation between the speed of journal and the coefficient of friction. In his tests upon lubricating oils, Beauchamp Tower showed that for a given oil tested at a constant temperature, the coefficient of friction, where there was perfect lubrication, varies directly as the square root of the surface speed of the journal, and inversely as the pressure per square inch of projected area. That is

$$\phi = K \frac{\sqrt{S}}{w} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [1]$$

S = surface speed of journal, feet per minute

w = load on bearing, pounds per square inch of projected area

TABLE 1 ABSTRACTS FROM LOG TESTS OF *B* GREASES NO. 5 DENSITY

Load on Bearing, 74.1 Lb. per Sq. In.

Time	Spring Balance				Bearing Temperature, Deg. Fahr.	Coefficient of Friction ϕ	Remarks
	P_1		P_2				
	Lb.	Oz.	Lb.	Oz.			
11.33	*7	8	....	....	76	0.0244	Readings marked *were taken immediately after forcing grease into the bearing
11.35	7	9	....	....	79	0.0276	
11.36	....	....	....	....	....	Reversed motor	
11.38	....	....	*6	9	80	0.0244	
11.39	....	....	6	8	80	0.0276	
11.40	....	....	6	7	80	0.0310	
11.40	....	....	*6	9	81	0.0244	
11.46	....	....	6	7.5	82	0.0293	
11.47	....	....	6	7	82	0.0310	
11.47	....	....	*6	11	82	0.0210	
11.57	....	....	6	5.25	83	0.0396	
11.57	....	....	*6	11.5	....	0.0201	
12.00	7	11.5	....	....	....	0.0325	
12.02	7	12.5	....	....	82	0.0350	
12.02	*7	8	....	....	....	0.0210	

Second Test at Same Load, but at Higher Temperatures

Time	P_1 Lb.	P_1 Oz.	P_2 Lb.	P_2 Oz.	Bearing Temperature, Deg. Fahr.	Coefficient of Friction ϕ	Remarks
5.26	7	6.5	....	....	110	0.0178	
5.26	*7	5.5	....	....	....	0.0146	
5.29	7	6	....	....	118	0.0162	
5.29	*7	5.5	....	....	....	0.0146	
5.30	....	....	6	12	120	0.0162	
5.32	....	....	6	12	....	0.0162	
5.32	....	....	*6	12.5	....	0.0146	
5.40	....	....	6	10	160	0.0228	
5.40	....	....	*6	10	160	0.0228	
5.42	7	8	....	....	164	0.0236	
5.43	7	7.5	....	....	160	0.0220	
5.46	7	8	....	....	158	0.0236	
5.47	8	0	....	....	....	0.049	Unsteady
5.47	*7	7.5	....	....	....	0.0220	
5.52	*7	10	....	....	185	0.026	
5.53	7	10	....	....	190	0.0260	
5.54	7	9.75	....	....	....	....	
5.55	....	....	6	10	....	0.0260	
5.55	....	....	*6	10	190	0.0260	
6.01	....	....	6	10	....	....	

K = coefficient, having different values for different lubricants, and for the same lubricant at different temperatures

From the observed values of S, w and ϕ in the tests of series A and B, the values of K were computed by substituting in equation [1], and were plotted against the corresponding temperature of the bearing. Fig. 10 shows the results of these plots for the X greases. Greases of different densities and series are distinguished by different forms of mark, so that the table from which each point came may be identified. Curves AB

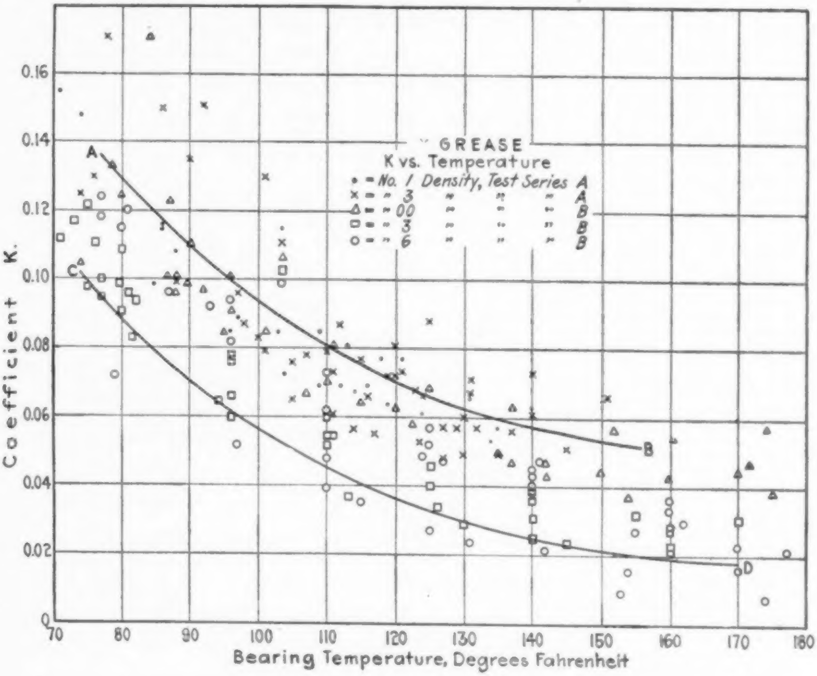


FIG. 10 BEARING TEMPERATURE AND COEFFICIENT OF FRICTION CURVE

and CD bound the areas above and below and represent the extreme values of K. It will be noticed that the plots for the denser greases in general, lie towards the upper curves, CD; and the softer ones nearer the curves AB.

27 It was desired to find the relation between K and the bearing temperature, expressed in the form of an equation which might be of general application. For each of the four curves, the points were replotted on logarithmic cross-section paper. Fig. 11. It appears from an inspection of the figure that K may

be expressed in terms of the temperature by an equation of the form

$$K = \frac{M}{t^n}$$

where M = a constant; t = bearing temperature, deg. fahr.; n =

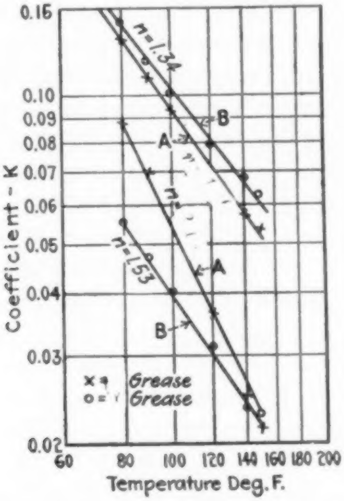


FIG. 11 K-BEARING TEMPERATURE CURVES PLOTTED ON LOGARITHMIC CROSS-SECTION PAPER

an exponent. The following values for these numbers are found from the logarithmic plots:

$$\text{Greases } X\text{, curve } AB \quad K = \frac{78}{t^{1.46}}$$

$$\text{curve } CD \quad K = \frac{1370}{t^{2.19}}$$

$$\text{Greases } Y\text{, curve } AB \quad K = \frac{49}{t^{1.34}}$$

$$\text{curve } CD \quad K = \frac{42}{t^{1.53}}$$

TESTS OF GREASE DENSITY

28 Lubricating oils are compared by certain physical tests. Prominent among these is the test for viscosity which is determined by a viscosimeter, several forms being in common use. One consists of a vessel for the oil surrounded by a space which

may be filled with water for the purpose of controlling the temperature. At the bottom of the vessel is a nozzle, through which a measured quantity of oil is permitted to flow. The viscosity is proportional to the time required to flow through the nozzle.

29 It seemed to the author that a similar scheme might be employed for comparing greases as to their consistency or density. Since grease is a solid and will not flow of itself, some compulsion must be used to force the grease through the nozzle. After preliminary experimentation an apparatus, shown in Fig. 7, was constructed which consisted of an ordinary grease cup supported upon an iron framework. A plunger was made to fit the cup, and to insure perfect freedom of motion the plunger was made slightly spherical. The plunger rod was carried through a guide, and supported weights placed upon its upper end. A scale graduated in twentieths of an inch was scribed on the rod, so that the time of descent over a measured distance might be noted. A nozzle of about 1/4 in. in diameter was placed in the bottom of the cup.

30 Experiments with this instrument gave results that were decidedly surprising. It was found that the density of a given grease, as indicated by this means, is a very variable quality. Successive passages of the same grease gave constantly decreasing lengths of time for the same distance. The grease became softer and more fluid by the process of forcing it through the nozzle. This was particularly true of the harder greases. After several passages, the grease becomes oily in appearance. The change may be due to a more thorough mixing of the ingredients composing the grease. The results of a number of tests with this instrument follow:

DENSITY TEST, *X* GREASE, NO. 1 DENSITY

31 The same grease sample was passed repeatedly through the nozzle. The weight on the plunger was 20 lb.; temperature of grease, 71 deg.

No.	1	2	3	4	5	6	7	8	9	10	11	12	13
Seconds to descend 1 in.	3750	235	95	66.2	57.4	36	27.2	23	19	19	16	24	14

The time became nearly constant after eight passages; the mean of Nos. 9, 10, 11, 12, 13 is 18.2 seconds.

32 The second test was made several days later, on the same grease sample as the preceding; load on plunger, 20 lb.; temperature of grease, 68 to 72 deg.

No.	1	2	3	4	5	
Seconds to descend 1 in.	17	12.6	14.2	13.8	14	mean time of last four, 13.6

33 The load was changed to 10 lb. and continued on the same grease sample as above; temperature of grease, 62 to 65 deg.

No.	1	2	3	4	
Seconds to descend 1 in.	572	670	597	593	mean time, 608

DENSITY TEST, *Y* GREASE, NO. 3 DENSITY

34 A sample of the grease was passed through the nozzle 18 successive times with weights of 20, 15 and 10 lb. The temperature of the grease was 82 deg. at the start, increasing to 92 deg. at the end.

LOAD, 20 LB.

No.	1	2	3	4	5	6	7	
Seconds to descend 1 in.	50	17.4	13	3.4 (?)	6.4	6.8	5.2	mean time of last four, 6.1

LOAD, 15 LB.

No.	8	9	10	11	12	13	
Seconds to descend 1 in.	15.8	16	12	11.4	11.6	9.6	mean time of last four, 11.1

LOAD, 10 LB.

No.	14	15	16	17	18
Seconds to descend 1 in.	159	131	129.6	97.8	95

DENSITY TEST, GREASE SERIES *B*, NO. 1

No.	1	2	3	4	5	6	7	8	9	10	11	12	13
Load, lb.	10	5	5	5	5	5	5	5	4	4	3	3	3
Seconds to descend 1 in.	3.2	23.6	31	18	27.4	25	23.8	19.8	57	59.4	273	250	267

35 Tests Nos. 2, 3, 4 and 5 were made on successive samples of grease taken from the can. Nos. 6 and 7 were repetitions of grease that had been passed through once. Similarly, Nos. 11 and 12 were new grease, while No. 13 was the second passage. It will be noted that, for this very thin grease, no great change occurs with successive repetitions of the test on the same sample. The time for No. 6 is almost exactly the same as the mean of the preceding four; and No. 13 is close to the mean of Nos. 11 and 12.

36 The consistency of grease as shown by the experiments described above becomes nearly constant after several passages through the apparatus at a constant load; but it appears that when the load is decreased, the grease again requires a number of passages under the new condition before coming to a constant condition of consistency. It is interesting to note the great effect

produced in the time of flow of the grease by a small change in the weight. Thus, in the last test, *Y* grease, No. 1, the time was increased about 450 per cent by decreasing the load from 4 to 3 lb.

CONCLUSIONS

37 Grease lubrication compares favorably with oil where the form of bearing is such as to favor the retention of the film of lubricant, and provision is made for an ample supply to the bearing. But, as shown by the experiments of series *A*, oil will give better results in case of a short bearing in proportion to the diameter.

38 Grease of soft consistency is a much better lubricant than the harder densities of the same grease. The advantage of the softer grease is especially marked at low temperatures, such as usually obtain in a well lubricated bearing.

39 The best method of applying grease to a bearing is by forced feed and a constant rate of flow. This agrees with the best practice in oil lubrication where the bearing is flooded with oil, which passes to a filter and is used again. The drawback in case of grease is the cleaning it after it has once passed through the bearing so that it can be used again. Intermittent application of grease means irregularity in the value of the coefficient of friction.

40 Grease cups with spring actuated plungers are designed to give a constant flow of grease. They are far from accomplishing this purpose, however. When such a cup is full of grease, the spring is compressed to its fullest amount, and the pressure upon the grease is, therefore, much greater than when the cup is nearly empty. Provision is made to regulate the flow by means of a small cock placed in the outlet of the cup, but this needs adjustment as the cup empties, and is apt to be neglected. The experiments upon grease consistency show what a great difference in flow is produced by a small change in the pressure upon the grease. A design of cup is desirable which will deliver the grease at a constant rate from the time it is filled until it is empty.

APPENDIX

TABLE 2 *X* GREASE, NO. 3 DENSITY

SERIES *B:* LOADS OF 53.5 TO 153.8 LB. PER SQ. IN.; DIAMETER OF JOURNAL 1¼ IN., LENGTH 2 IN.; GREASE CUP USED WITH SPRING ACTUATED PLUNGER

Number	Bearing Temperature, Deg. Fahr.	Coefficient of Friction	Remarks
Load on Bearing 53.5 Lb. per Sq. In.; Surface Speed of Journal 405 Ft. per Min.			
1	76	0.0414	Room temperature 72 deg.
2	80	0.0342	
3	81	0.0358	
4	82	0.0352	
5	92–96	0.0268	
6	94–97	0.0246	
7	95	0.0218	
8	110–114	0.0190	
9	111	0.0213	
10	111	0.0213	
11	124	0.0173	
12	125	0.0185	
13	126	0.0168	
14	141	0.0145	
15	141	0.0134	
16	140	0.0151	
17	154	0.0134	
18	156	0.0112	
19	157	0.0117	
20	170	0.0101	
21	168	0.0106	
22	170	0.0140	
Load on Bearing 74 Lb. per Sq. In.; Surface Speed of Journal 395 Ft. per Min.			
1	78	0.0276	
2	79	0.0252	
3	80	0.0244	
4	95	0.0187	
5	96	0.0199	
6	97	0.0211	
7	120	0.0122	
8	119–122	0.0134	
9	120	0.0097	
10	135	0.00813	
11	134	0.0069	
12	135	0.0065	
13	152	0.0049	
14	155	0.0049	
15	155	0.00406	
16	168–171	0.0041	
17	171	0.0053	
18	171–189	0.0057	

TABLE 2—(*Continued*)

Number	Bearing Temperature, Deg. Fahr.	Coefficient of Friction	Remarks
Load on Bearing 94.1 Lb. per Sq. In.; Surface Speed of Journal 380 Ft. per Min.			
1	61	0.0312	Room temperature 67 deg.
2	69	0.0262	
3	73	0.0242	
4	75	0.0204	
5	76	0.0198	
6	96	0.0131	
7	95	0.0121	
8	95	0.0124	
9	110	0.0086	
10	111	0.0115	
11	100–110	0.0127	
12	124	0.0079	
13	126	0.0064	
14	125	0.0067	
15	143	0.0048	
16	141	0.0054	
17	139	0.0054	
18	161	0.0045	
19	160	0.0042	
20	160	0.0048	
Load on Bearing 114.2 Lb. per Sq. In.; Surface Speed of Journal 390 Ft. per Min.			
1	75	0.0212	Room temperature 67 deg.
2	77	0.0173	
3	78	0.0171	
4	79	0.0171	
5	96	0.0145	
6	96	0.0137	
7	96	0.0123	
8	108–111	0.0123	
9	112	0.0094	
10	110	0.0094	
11	127	0.0073	
12	125	0.0068	
13	125	0.0068	
14	140	0.0059	
15	140	0.0052	
16	140	0.0052	
17	159	0.0039	
18	161	0.0037	
19	160	0.0039	

TABLE 2—(*Continued*)

Number	Bearing Temperature, Deg. Fahr.	Coefficient of Friction	Remarks
Load on Bearing 133.4 Lb. per Sq. In.; Surface Speed of Journal 390 Ft. per Min.			
1	80	0.0162	
2	83	0.0119	
3	80	0.0126	
Load on Bearing 133.4 Lb. per Sq. In.; Surface Speed of Journal 390 Ft. per Min.			
4	97	0.0090	
5	96	0.0104	
6	96	0.0099	
7	114	0.0061	
8	113	0.0052	
9	113	0.0052	
10	131	0.0049	
11	130	0.0043	
12	132	0.0038	
13	144	0.0038	
14	146	0.0034	
15	145	0.0034	
16	158–162	0.0038	
17	161	0.0043	
18	161	0.0043	
Load on Bearing 153.8 Lb. per Sq. In.; Surface Speed of Journal 390 Ft. per Min.			
1	56	0.0285	Room temperature 62 deg.
2	60	0.0223	
3	64	0.0156	
4	67	0.0148	
5	71	0.0145	
6	77	0.0121	
7	93	0.0098	
8	94	0.0103	
9	95	0.0090	
10	108	0.0074	
11	108	0.0070	
12	110	0.0070	
13	123	0.0060	
14	125	0.0055	
15	124	0.0059	
16	139	0.0049	
17	140	0.0047	
18	141	0.0041	
19	158	0.0035	
20	160	0.0037	
21	160	0.0037	

TABLE 3 Y GREASE, NO. 2 DENSITY

SERIES A: LOADS OF 46 TO 85 LB. PER SQ. IN.; DIAMETER OF JOURNAL 2⅝ IN., LENGTH 1½ IN.; COMPRESSION GREASE CUP USED WITH INTERMITTENT FEED OF THE LUBRICANT

Number	Bearing Temperature, Deg. Fahr.	Coefficient of Friction	Remarks
Load on Bearing 46 Lb. per Sq. In.; Surface Speed of Journal 800 Ft. per Min.			
1	70	0.0504	
2	80	0.0460	
3	82	0.0421	
4	90	0.0400	
5	99	0.0340	
6	123	0.0355	
7	140	0.0378	
8	180	0.0300	
Load on Bearing 72 Lb. per Sq. In.; Surface Speed of Journal 725 to 800 Ft. per Min.			
1	69	0.0365	
2	71	0.0330	
3	83	0.0294	
4	95	0.0292	
5	115	0.0240	
6	147	0.0252	
7	208	0.0232	
Load on Bearing 85 Lb. per Sq. In.; Surface Speed of Journal 700 Ft. per Min.			
1	70	0.0350	
2	80	0.0282	
3	90	0.0254	
4	100	0.0247	
5	126	0.0183	

TABLE 4 ENGINE OIL

SERIES A: LOADS OF 20 TO 174 LB. PER SQ. IN.; DIAMETER OF BEARING 2¾ IN., LENGTH 1½ IN.; TESTS OF THIS OIL WERE FOR THE PURPOSE OF COMPARING OIL WITH GREASE UNDER THE SAME CONDITIONS

Number	Bearing Temperature, Deg. Fahr.	Coefficient of Friction	Remarks
Load on Bearing 20 Lb. per Sq. In.; Surface Speed of Journal 825 Ft. per Min.			
1	79	0.0630	Room temperature 79 deg.
2	82	0.0541	Motor was reversed at each observation
3	87	0.0470	
4	90	0.0455	
5	92	0.0435	
6	96	0.0405	
7	100	0.0399	
8	105	0.0362	
9	108	0.0344	
10	109	0.0389	
11	110	0.0380	
12	111	0.0330	
13	111	0.0330	
14	111	0.0330	
Load on Bearing 46 Lb. per Sq. In.; Surface Speed of Journal 825 Ft. per Min.			
1	88	0.0292	Room temperature 90 deg.
2	93	0.0236	Motor was reversed at each observation
3	98	0.0222	
4	101	0.0206	
5	104	0.0200	
6	106	0.0198	
7	108	0.0186	
8	109	0.0186	
9	111	0.0182	
10	112	0.0190	
11	114	0.0176	
12	115	0.0178	
13	116	0.0178	
14	117	0.0178	
15	118	0.0176	
16	...	0.0178	
17	119	0.0176	
18	120	0.0176	
Load on the Bearing 71.5 Lb. per Sq. In.; Surface Speed of Journal 825 Ft. per Min.			
1	81	0.0234	Motor was reversed at each observation.
2	86	0.0209	
3	92	0.0173	
4	96	0.0170	
5	102	0.0170	
6	107	0.0146	
7	109	0.0142	
8	110	0.0142	
9	111	0.0135	
10	112	0.0135	
11	112	0.0130	
12	113	0.0130	
13	113	0.0136	
14	115	0.0136	
15	115	0.0137	
16	116	0.0129	

TABLE 4—(*Continued*)

Number	Bearing Temperature, Deg. Fahr.	Coefficient of Friction	Remarks
Load on Bearing 97 Lb. per Sq. In.; Surface Speed of Journal 690 to 825 Ft. per Min.			
1	89	0.0138	Room temperature 90 deg.
2	96	0.0137	Motor was reversed at each observation.
3	101	0.0133	
4	107	0.0129	
5	107	0.0123	
6	111	0.0121	
7	114	0.0119	
8	116	0.0120	
9	118	0.0120	
10	119	0.0120	
11	120	0.0120	
12	122	0.0119	
13	123	0.0119	
14	124	0.0117	
15	...	0.0121	
16	127	0.0117	
17	127	0.0105	
18	128	0.0105	
19	128	0.0105	
20	129	0.0104	
21	129	0.0105	
Load on Bearing 122 Lb. per Sq. In.; Surface Speed of Journal 825 Ft. per Min.			
1	75	0.0196	Room temperature 77 deg.
2	83	0.0144	
3	89	0.0136	
4	94	0.0099	
5	98	0.0126	Motor reversed
6	102	0.0113	
7	104	0.0108	
8	...	0.0113	Motor reversed
9	106	0.0107	
10	108	0.0099	
11	110	0.0099	
12	...	0.0095	Motor reversed
13	111	0.0092	
14	...	0.0099	
15	113	0.0107	
16	...	0.0077	Motor reversed
17	115	0.0101	

TABLE 4—(*Continued*)

Number	Bearing Temperature, Deg. Fahr.	Coefficient of Friction	Remarks
Same load and speed; second test			
1	81	0.0151	
2	87	0.0143	Motor reversed
3	95	0.0131	Motor reversed
4	101	0.0114	Motor reversed
5	104	0.0109	Motor reversed
6	106	0.0109	
7	109	0.0110	Motor reversed
8	111	0.0108	Motor reversed
9	113	0.0108	
10	115	0.0109	Motor reversed
11	116	0.0107	Motor reversed
12	118	0.0106	
13	119	0.0106	Motor reversed
14	121	0.0095	
15	123	0.0091	Motor reversed
16	124	0.0098	Motor reversed
17	124	0.0098	
18	...	0.0096	
Load on Bearing 148 Lb. per Sq. In.; Surface Speed of Journal 825 Ft. per Min.			
1	75	0.0172	Room temperature 76 deg.
2	82	0.0142	Motor was reversed at each observation
3	90	0.0120	
4	96	0.0108	
5	101	0.0108	
6	105	0.0113	
7	110	0.0121	
8	113	0.0105	
9	114	0.0094	
10	116	0.0083	
11	118	0.0082	
12	120	0.0082	
Load on Bearing 174 Lb. per Sq. In.; Surface Speed of Journal 755 to 825 Ft. per Min.			
1	84	0.0134	Room temperature 80 deg.
2	91	0.0109	Motor was reversed at each observation
3	96	0.0100	
4	100	0.0101	
5	106	0.0101	
6	111	0.0098	
7	114	0.0082	
8	116	0.0061	
9	117	0.0060	
10	119	0.0061	
11	120	0.0060	
12	121	0.0059	

FOREIGN REVIEW

BRIEF ABSTRACTS OF CURRENT ARTICLES IN FOREIGN PERIODICALS

CONTENTS

The Editor will be pleased to receive inquiries for further information in connection with articles reported in the Review. Articles are classified as *c* comparative; *d* descriptive; *e* experimental; *g* general; *h* historical; *m* mathematical; *p* practical; *s* statistical; *t* theoretical. Articles of exceptional merit are rated *A* by the reviewer. Opinions expressed are those of the reviewer, not of the Society.

FOREIGN REVIEW

Through the collaboration of the Library Committee of the United Engineering Societies, the range of papers available for the Foreign Review is gradually being extended so as to cover all the most important non-English publications. In the present issue, for instance, several articles are abstracted from Swedish and Russian periodicals. It is the intention of the Editor to give brief abstracts of articles even of secondary importance from these publications, because otherwise they would most likely remain entirely unknown to the engineering profession, the publications containing them, as far as the Editor is aware, not being generally covered by engineering indexes.

THIS MONTH'S ARTICLES

An abstract of the article by Kaplan on the two dimensional water turbine theory is given: a runner constructed in accordance with this theory is said to have shown an efficiency of 80 per cent. Gas power men will be interested in the description of the Marischka combined gas producer and boiler, with its arrangement for fully utilizing the heat of the gases. In the same section is described the slide valve gear of the Dubois-Rousseau engine, which is of interest owing to its apparent simplicity; another engine, described in the Russian technical paper *Dvigatel*, represents an attempt to construct a Diesel engine without any fuel atomizing devices, and moreover requiring no lubrication, part of the fuel used acting as a lubricant, and then burning on a special cycle of its own. Data of the tests on Diesel-electric locomotives on the Swedish state railroads are also given, the conditions of operation of the locomotives under tests corresponding somewhat to those prevailing in the suburban traffic of the larger American cities. In the section Mechanics attention is called to the article describing the method for the experimental determination of the moment of inertia of runners, permitting one to judge, to a certain extent, of the homogeneity of the materials used, a question sometimes of great importance in rotors of large dimensions and subject to considerable mechanical stresses. In this connection may also be mentioned an article on alloys for use in high-pressure steam

turbines, giving data of the work done in this respect by German government laboratories and manufacturers. R. Slaby's brief article indicates a simple way of constructing a differential curve to any given one, without having to determine tangents, and using the simplest graphical operations and tools. Owing to lack of space, the part of the article in which Mr. Slaby shows mathematically that his curve is not an approximation, but a real differential curve, had to be omitted. The brief notice by Professor Timiriazeff on the internal friction of rarefied gases, though classed under Mechanics, may also be of interest to the gas and gasolene engine designers, as bearing on the carbureter and gas mixer construction.

The Schuch rivetting indicator is briefly described: it shows to the workman how long the pressure on the rivet is kept, and at the same time makes automatically records valuable both for the determination of the quality of the job done, and for the accounting and rate setting departments.

In the section Steam Engineering is described the Breguet Ejectair, a new and apparently efficient apparatus for producing vacuum in condensers of steam engines. The principle of the apparatus is not new, but the design has several interesting features. The Lomshakoff furnace is illustrated, and mention is made that in tests with it difficulty was experienced owing to lack of apparatus, showing the efficiency of furnaces with undergrate draft, as well as the general usefulness of using compressed air for undergrate draft. The next abstract gives data of tests with the Marcotty smoke consuming apparatus on locomotives, proving the economy of installing such a device.

Several articles on the elastic limits of alloys, distribution of stresses in notched bars under tension, expansion of nickel steel at temperatures up to 300 deg. cent. (572 deg. fahr.), sound conductivity of building materials and walls, etc., are also given. It is proposed to start a new section devoted to abstracts of articles on railroad matters in one of the early issues.

Hydraulics

Two-Dimensional Turbine Theory, with a Consideration of Water Friction, and Its Application to the Design of Blades (*Die zweidimensionale Turbinentheorie mit Berücksichtigung der Wasserreibung und ihre Anwendung und Ergebnisse bei Schaufelkonstruktionen*, V. Kaplan. *Zeits. für das gesante Turbinenwesen. Serial article.*

The main difficulty of a theoretical treatment of flow processes lies in an appropriate consideration of internal liquid friction and wall friction. The author has previously established that, owing to the viscosity of the fluid,

in each fluid flow there is a tendency for the establishment of a state of resistance characterized by the fact that in a section normal to the direction of flow all units (by weight) of the flowing liquid contain the same amount of energy. The author calls a flow in this state of resistance "free flow." The wall friction in this case need not be considered. Should a working flow, i.e. one that besides effecting its own motion delivers work to the outside, be "free," a uniform amount of energy must be taken off from every section of the flow. When, however, no work is delivered outwards, the flowing system is at rest, and the author calls it a "no-work flow." The purely analytical investigation of a flow is especially difficult when the shape of the passage is difficult to express analytically, as is usually the case with the water passages in turbine runners. Whenever available, graphical results are to be preferred on account of their being easier to comprehend and simpler in expression. For a free flow pressure and velocity at any particular spot in all practical cases may be expressed in two-dimensional flow images, provided the wall friction is neglected and

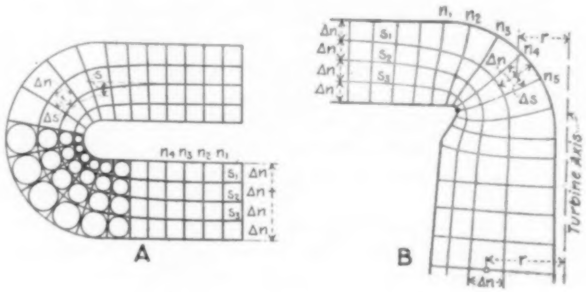


Fig. 1 Flow Images, Water Turbine Wheel

certain conditions simplified. As an example may be shown (Fig. 1A) the flow image of a U-shaped passage. The total flow is divided into partial flows of equal rate, and, since the energy contents in all the partial flows in a cross-section normal to the direction of flow have been shown to be constantly equal to one another, along such a section the relation must exist of $\frac{\triangle n}{\triangle S} = \text{const.}$ It is convenient to select $\frac{\triangle n}{\triangle S} = 1$, in which case the construction may be facilitated by the little circles shown in the Fig. 1A. In the flow image the stream lines $s_1, s_2, s_3 \ldots$, indicate the limits of partial streams of equal water delivery per second, while the lines normal to them indicate sections of equal energy content ("lines of level"). For a free flow in a hollow body of revolution, such as occurs in guide wheels and runners of a Francis turbine, Fig. 1B, there is a further condition that $r. \frac{\triangle n}{\triangle S}$ must be constant. In accordance with this flow image a blade with no wall friction may be constructed, as set full in the article (for practical purposes a correction for wall friction must be made, however).

The following has been found with respect to flow accompanied by friction: If c_1 is the velocity of flow in a canal with frictionless walls, it is reduced to $c = \frac{c_1}{k_t}$ under the action of bottom friction, and to $c = \frac{c_1}{k_s}$ by the side wall friction. Since, however, the bottom and side wall frictions act simultaneously, strictly speaking they may be represented only in three dimensions. The coefficient k depends on the roughness of the surfaces and their distance from one another, and varies from infinity for distance zero to 1 for distance ∞. Experiments have shown however that the value 1 is actually reached at comparatively small distance between the walls (about 40 mm, or 1.6 in.). Care must be taken properly to consider the action of strong curvatures; thus, in Fig. 1B with free flow the maximum velocity of flow appears to be at A, while the actual velocity at that point is 0. The design of turbine runners with respect to wall friction has naturally to depend on certain assumptions subject to subsequent correction, due to the fact that the resistances are affected by the type of body design which has still to be found. The design fully described by the author is characterized by the fact that the lines of flow are pressed somewhat close to the middle line of flow, due again to the fact that the velocities nearer the passage walls are made somewhat smaller.

To test the correctness of the above theory, a runner 100 mm (4 in.) in diameter has been constructed, and compared with one designed in accordance with the usual unidimensional theory. The efficiencies found were 80 and 73 per cent in favor of the new type.

Prime Movers in Waterworks and their Influence on the Economic Diameter of Pressure Piping (*Über Antriebsarten von Pumpwerken und deren Einfluss auf den wirtschaftlichen Durchmesser von Druckrohrleitungen*, E. Rutsatz. *Journal für Gasbeleuchtung*, vol. 56, no. 19, p. 444, 8 pp., 6 figs. *c*). Discussion of comparative advantages of steam and internal-combustion engines and electric motors as prime-movers for water works. Nothing definite is said as to the relation between the kind of prime-mover and piping diameter.

Experiments on Pressure Variation in the Piping of a Francis Turbine Plant with Change of Load (*Versuche über die Druckänderungen in der Rohrleitung einer Francis-Turbinenanlage bei Belastungsänderungen*, A. Watzinger and O. Nissen. *Mitteilungen über Forschungsarbeiten auf dem Gebiete des Ingenieurwesens*, no. 134, 1913, p. 27, 18 pp., 34 figs. *e*). Experimental data, not suitable for abstracting: regulation phenomena and pressure variations with change of load investigated.

Rational Construction of Concrete Suction Pipes (*Construction rationelle des tuyaux d'aspiration en Béton*, Th. Kach. *La Houille blanche*, vol. 12, no. 4, p. 116, 2 pp., 3 figs. *t*). Brief exposition of the Dubs process for the determination of the shape of suction pipe in water turbine installations involving the least losses through residual velocity of the water (this is especially important with low heads, where the relative amount of power lost on this account is particularly large).

Internal Combustion Engines

Engines at the General Agricultural Exhibition (*Les machines au Concours General Agricole*, H. Pillaud. *La Technique moderne*, vol. 6, no.

10, p. 387. Serial article, *d*). Among other things, the article describes internal-combustion engines shown at the French General Agricultural Exhibition. The Dubois-Rousseau factory has shown a slide-valve engine of great simplicity, with an ordinary cylinder, and characterized only by a special valve gear located at the top of the cylinder normally to its axis. This valve gear is tube-shaped, with a section in the middle cut out so as to form a longitudinal port opening a communication with the cylinder. The two extremities have been slit along the generating line in a manner such that the cylinder might, under the action of pressure, open slightly and so make a better contact with the wall enclosing it. In its different positions, the valve gear may be in communication either with the inside of the cylinder, or with the admission or exhaust pipes; it can oscillate under the action of a single connecting rod driven from a cam running

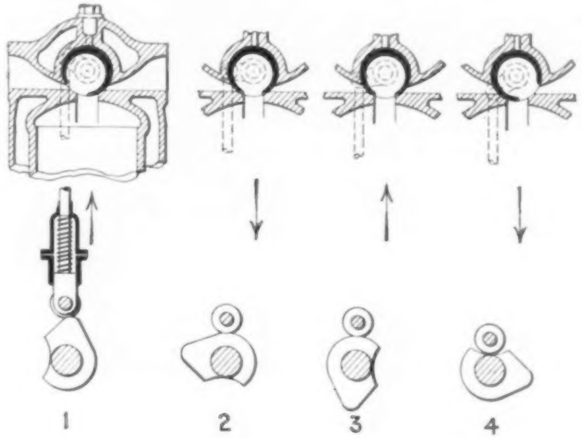


Fig. 2 Dubois-Rousseau Slide Valve Gear

at half the speed of the engine. Fig. 2 shows the position of the valve gear during the entire cycle. The valve gear is stationary during the second and third stroke, and it is then that its elasticity is of importance in permitting it to make a tight contact with the enclosing walls, and thus prevent leaks. The same valve gear may be applied also to multicylinder engines.

Concerning Crude Oil Engines (*Über Rohölmotoren*, F. Weinreb. *Elektrotechnik und Maschinenbau*, vol 31, no. 21, p. 444, May 25, 1913. 6 pp., 11 figs. *cdh*). A general article on the introduction and use of internal-combustion engines, and their economy. The paper contains numerous data showing that, in Germany and Russia, it has been found that for central station work the Diesel engine proved to be not only more reliable than the steam turbine, but also more economical. The greater reliability of the Diesel engine lies mainly in the fact that it permits a larger reserve storage of fuel than steam plants, and makes the plant better protected against interruptions of service caused by tie-ups in transportation or strikes.

Diesel-Electric Motor Cars (*Diesel-elektriske motorvogne, Elektroteknisk Tidsskrift*, vol. 26, no. 13, p. 99, May 5, 1913, 3 pp., 3 figs. *d*). In *The Journal* (May 1913, p. 888-889) reference was made to the Diesel-electric locomotives installed lately on the Swedish railroads. The present article describes them in detail. The locomotives are designed to run at 60 km (37.2 miles) maximum speed per hour, and take care of two trailers, of total weight 30 tons (33 short tons) besides that of passengers. The power equipment consists of a 6-cylinder, 75-effective-horsepower Diesel engine, running at 700 r.p.m., and a direct-connected 50 kw., 440 volt, direct current generator, supplying current to two 30-h.p. motors. The Diesel engine is of the four-stroke cycle type, with six working and one pump cylinder supplying compressed air for starting the engine and fuel injection. The air for the brakes is supplied from a separate little pump. The fuel tank is designed to hold fuel for a run of 1200 km (745 miles) with two trailers, and 1800 km (1120 miles) without the trailers.

The locomotive was tested in May 1912 in runs between the stations Enköping and Heby, distance 37 km (23 miles), containing grades as stiff as 11 promill, and minimum radii of curvature 300 mm (984 ft.). Between these two stations there are five intermediate stations at which stops were always made. The runs were made either with the motor car alone, or with trailers, passenger or goods cars, the weight of the trailers not exceeding 45 tons (49.5 short tons). The following data as to fuel (crude oil) consumption are given:

a Motor car alone, weight of train 26.5 tons (29.150 short tons), speed 40 to 45 km (24.8 to 27.9 miles) per hour; fuel consumption per run 8.8 kg (18.4 lb.), per train-kilometer 0.238 kg (0.842 lb. per train-mile).

b Motor car and trailer (passenger car), weight of train 40 tons (44 short tons), speed about 40 km (24.8 miles) per hour; fuel consumption per run 11.25 kg (24.75 lb.), per train-kilometer 0.304 kg (1.07 lb. per train-mile).

c Motor car, one passenger car, and one or two goods cars, train weight about 60 tons, speed 35 km (21.7 miles) per hour; fuel consumption per run 12.6 kg (27.72 lb.), per train-kilometer 0.342 kg (1.21 lb. per train mile).

Utilization of Blast Furnace Gases in Gas Engines (*Sur l'utilisation des gaz de hauts fourneaux dans le moteurs à gaz*, Ch. Reignier, *Société industrielle de L'Est*, no. 109, p. 21, April 1913. 10 pp., 2 figs. *dgh*). General, partly historical paper on the use of blast furnace gases in gas engines. The author asserts that a kw-hr. can be produced from blast furnace gases at two centimes (say $0.004), this figure being considered rather as a maximum for a well appointed plant.

Gas Producers for Power Gas (*Über Gaserzeuger für Kraftgas*, Gwosdz. *Oel- und Gasmaschine*, vol. 13, no. 2. Serial. *d*). Semi-annual account of progress in the field of gas producer engineering. In Fig. 3 is shown a *combined revolving grate producer and steam boiler* designed by the chief engineer of the Vienna Municipal Gas Works Marischka. The shaft of this producer, to its full length, is made into a steam boiler, divided into two parts with a large number of water tubes between, in order to utilize the heat contained in the gases leaving the producer. In addition the larger part of the boiler is provided with a jacket, which forces the gases to flow

along the surface of the boiler during a considerable part of their way out. By this arrangement it has proved possible to reduce the temperature of the gases leaving the producer to 220 deg. cent. (428 deg. fahr.), while the steam in the boiler is brought to six atmospheres gage pressure. Only part of this steam is required for the gas producer itself; the rest may be used for general requirements. The chief value of this producer lies in the fact that it takes care of cooling the gases down to near the temperature at which they can be conveniently handled in the gas engine, and at the same time utilizes their excess of heat. This is not the first attempt to combine a gas producer with a boiler, but it is only the revolving grate that made such a construction commercially practicable, owing to the large

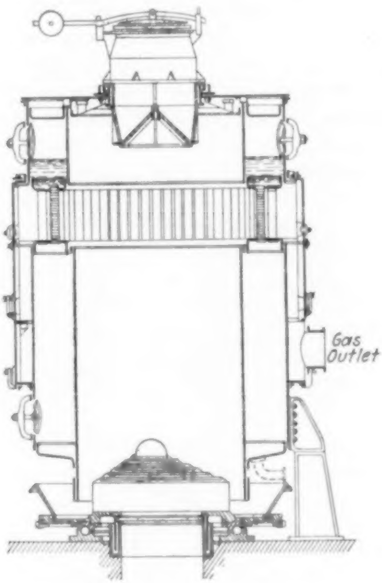


FIG. 3 MARISCHKA COMBINED REVOLVING GRATE PRODUCER AND STEAM BOILER

amount of fuel handled, and correspondingly large amount of heat developed per unit of shaft cross-section. Attempts to use the same construction in gas producers with by-product recovery do not appear to have been successful, owing to settling of tar in the boiler jacket.

DELIVERY OF FUEL INTO THE CYLINDERS OF INTERNAL-COMBUSTION ENGINES FROM UNDER THE PISTON RINGS (*O podache garyoocheva v zylindry drigateley vnootrennyeva sgaraniya ispod porshnevykh kaletz*, V. Pokszïszevski. *Drigatel*, vol. 7, nos. 3 and 7, pp. 37 and 105. 7 pp., 6 figs. *d*). The author considers mainly engines of the Diesel type, and finds the following imperfections in their design: (*a*) the atomizer is a very delicate device, not easily adjustable, and apt to get out of adjustment with change in the consistency of fuel; further, it requires a high air compression, especially

when the engine is burning oil residues (mazout); the air compressor and atomizer are the two weak elements in the Diesel engine operation; (*b*) lubricating oil is a heavy expense, owing to the high cost of oil suitable for internal-combustion engines; the author states that, at the prices of oil and fuel prevailing in Russia, the lubrication costs from 13 to 17 per cent of the cost of fuel. His idea is therefore to construct an engine that would work without fuel atomizer devices, and provide its own lubrication. This is to be done in the following manner (only a brief description showing the principle of the new design is given here: the engine has not yet been built, and therefore it is not considered worth while to go extensively into details): A ring *C* (Fig. 4) shaped as shown, of some soft and elastic material, such as soft iron, bronze, etc., is provided on the piston, its diameter being about 0.02 in. less than that of the cylinder in which the piston moves. The fuel (naphtha or oil residues) passes from a tank (not shown) where it is periodically under pressure, through pipes *i* and *b* into the ring shaped space *a* under the ring *C*. The cycle of the engine is then as follows:

a forward stroke: working, combustion.

b backward stroke: exhaust of the products of combustion, accompanied, if desired, by air scavenging.

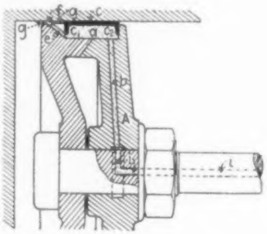


Fig. 4 Internal Combustion Engine Piston with Fuel Delivery through the Piston Rings

c forward stroke; air admission and exhaust valves are both closed; a vacuum is formed in the cylinder; at the same time the fuel in the space *a* under the ring *C* is set under pressure from the tank; as a result, while the piston moves forward, the fuel is forced out through the duct *e* into the passage *f*, whence it goes to cover by a thin layer the cylinder walls; the comparatively high temperature of the cylinder walls, together with the partial vacuum in the cylinder, combine in producing the evaporation of a large part of the fuel thus spread on the walls of the cylinder; the heaviest constituents of the naphtha will however not evaporate, and it is they that form a layer taking care of the cylinder lubrication. At the end of this stroke air is admitted into the cylinder, either direct from the atmosphere, or under some compression.

d backward stroke: the mixture of air and naphtha vapor is compressed and, when near the dead point, ignited, either artifically, or through its own compression.

$e =$ (*a*) forward stroke, working, combustion; the mixture of air and naphtha vapor burn on the Otto cycle, while the naphtha which stayed on the cylinder walls, and had no time or chance to evaporate, burns additionally, and thermally superimposes itself on the Otto cycle. The operation of the engine is therefore between the Otto and Diesel cycles.

There is of course a certain possibility of uncombustible residues of the naphtha used settling on the cylinder walls; as advantages of the described construction are pointed out by the author: reduction in the size and weight of the piston, and reduction in the consumption of cooling water, due to partial cooling produced by the spreading and evaporation of naphtha.

Machine Shop

Concerning the Basis for Determining the Power Consumption in Press Forging (*Über die Grundlagen zur Ermittlung des Arbeitsbedarfes beim Schmieden unter der Presse*, Fr. Riedel. *Zeits. des Vereines deutscher Ingenieure*, vol. 57, no. 22, p. 845, May 31, 1913. 6½ pp., 21 figs. *e*). The scope of the article is considerably broader than its title would indicate. The author points out that, while the power consumption in press forging may be determined, too little is known as yet to establish a law showing the relation between the amount of power consumed and the change of form produced thereby. He shows that such a relation is materially affected by the formation, when plastic bodies are compressed, of slip cones (*Rutschkegel*), and how actual data as to the strength of wrought iron at various temperatures may be obtained by means of an electric furnace. He discusses at some length the equations regulating cooling of iron by conduction and radiation.

Schuch Riveting Indicator (*Schuchscher Nietkontroller*, G. Hilliger. *Zeits. für Dampfkessel und Maschinenbetrieb*, vol. 36, no. 22, p. 263, May 30, 1913. 2½ pp., 5 figs. *d*). There has been no way to determine the quality of a riveting job besides knocking of some rivet heads, and inspecting the fractures. In the riveting process itself there have been established practically and experimentally certain lengths of time for the pressure to be applied on the rivet head, but there was no certainty that these were actually adhered to; rather the contrary is to be assumed to be more generally the case: the workman's attention is so much taken up by handling his tools that, with no stop watch at his command, he is hardly likely to be able to time his operation to within the seconds necessary to make a really good job of riveting. The Schuch riveting indicator is intended to obviate all these difficulties. It consists of two parts: The first records and indicates the pressure applied, and time during which it is applied, while the second simply draws a time line. When the riveting is started, the pressure-time line gradually rises until the full pressure is applied; the workman has then conveniently located in front of him a seconds indicator which permits him to keep the pressure for just as long as is wanted: The second line indicates the time of day when each rivet was made, and thus permits keeping account of the amount and steadiness of work done, giving valuable data for making up the pay check. The apparatus works

automatically, and appears to be of simple construction. No detailed description is here given because the illustrations in the original are somewhat blurred, and would be difficult to reproduce.

Measuring Instruments

SAFETY DEVICE FOR USE IN CONNECTION WITH THE JUNKERS RECORDING CALORIMETER (*Eine Sicherungsvorrichtung für das Junkers'sche Registrierkalorimeter*, W. Allner. *Journal für Gasbeleuchtung*, vol. 56, no. 19. p. 438, 3 pp., 4 figs. *d*). One of the conditions of correct working of the Junkers recording calorimeter is that the gas and water (the latter used for taking up the heat of combustion of the gas) should flow to the calorimeter uninterruptedly. The device described consists of an electrically operated cock which shuts off the gas admission to the calorimeter whenever there occurs a disturbance in the flow of either gas or water, and at the same time sounds an alarm clock. This device prevents both the possibility of overheating the calorimeter and formation of an explosive mixture inside it. The device is said to have been in use for nearly a year and given satisfaction.

Mechanics

EXPERIMENTAL DETERMINATION OF THE MOMENT OF INERTIA OF RUNNERS (*Experimentelle Bestimmung der Trägheitsmomente von Laufrädern*, A. Lechner. *Dinglers polytechnisches Journal*, vol. 328, no. 22, p. 337, May 31, 1913. 2½ pp., 2 figs. *e*). The moment of inertia of a body has been hitherto determined by the method of oscillations, or from the inertia period, or by means of the Atwood machine. The author proposes a new method based on the following principle. When a pair of wheels rolls down an inclined plane in a straight line, a condition of pure rolling is that, first, there is friction of adhesion, or that $(R) \leqq N.f$, and second that $v - r\ \ \omega = 0$, where v is the velocity of the center of gravity of the system, ω angular velocity, r radius of the wheel, N normal pressuré, and f coefficient of friction. If further, T be the moment of inertia, m mass of the system ψ angle of inclination of the plane to the horizontal, then:

$$m\frac{dx}{dt} = mg \sin \varphi - R \ldots\ldots [1]$$

and

$$0 = N - mg \cos \varphi \ldots\ldots [2]$$

$$T\frac{d\omega}{dt} = R \cdot r \ldots\ldots [3]$$

and since $v = r\omega$, it follows that

$$\frac{dv}{dt} = g \sin \varphi \frac{mr^2}{T + mr^2} \ldots\ldots [4]$$

The acceleration $\frac{dv}{dt}$ may be expressed by the length of the path s and respective time t, in which case [4] becomes

$$T = mr^2\left(g\frac{\sin \varphi}{2\,s} \cdot t^2 - 1\right) \ldots\ldots [5]$$

from which T may be determined. The author shows how he has proved by the use of a supplementary inequality that he has been dealing with pure rolling.

Fig. 5A shows the general arrangement of the testing apparatus. The plane AB is horizontal; the angle φ determined from the values of AC and AE. On the axis of the body subject to rotation is fixed a needle and the body itself is placed in its initial position at E in such a manner that the point of the needle is just covered by the vertically hanging thread; the body is kept in this position by a powerful electromagnet. A stop-watch is set into action at the moment when the current is interrupted in the electromagnet circuit at W, and is stopped when the needle touches the other vertical thread GD. The measurement of time is of the greatest importance in this process. In order to establish the moment of the passage of the needle by the thread GD, a telescope L was placed in front of the thread on which a ray of light was projected. In other tests the apparatus was arranged so that the observer could see only the points E and D. Several other methods were tried, but it was found that the more elaborate processes gave practically the same results as direct observation. Full data of the tests are given. The values obtained for the wheel tested varied

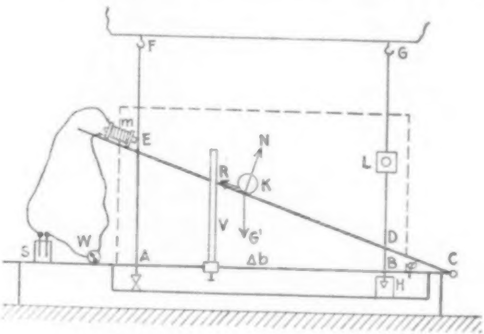


Fig. 5 Arrangement of Apparatus for Determining the Moment of Inertia of Runners

between 418 and 461 g cm^3, while the calculated value for the wheel under test was 436 g cm^3. It must be borne in mind, however, that the calculated value is based on the assumption of perfectly uniform distribution of matter in the structure, and no allowance is made for possible lack of homogeneity of the material, blowholes, etc. The method might be considerably improved by introducing automatic time recording, in which case it could be used for large wheels.

Approximation Method for the Investigation of Stability of Elastic Systems (*Näherungsmethode zur Untersuchung der Stabilität elastischer Systeme* S. Timoschenko. *Kiewer Univ.-Nachr.*, 52, no. 7, p. 25, 1912, through *Beiblätter zu den Annalen der Physik*, vol. 37, no. 9, p. 579, 1913. *et*). When one or two dimensions of an elastic body are very small, small deformations may produce large changes of form. Then, starting from a "critical" load, there may be several equilibria with a single system of forces, and the one corresponding to a minimum of potential energy is a stable one. With the critical load the work of external forces T required for displacement of the system from its equilibrium is equal to the change in the internal energy of the elastic system V, and an approximate

determination of the critical load may be effected by determining, with due restrictions as to the limiting conditions, some equilibrium of the values of T and V. This method gives somewhat high values for the critical load, and several determinations have to be made before the degree of approximation can be estimated. The original article shows how this method is applied to technical problems some of which have not yet been solved owing to the difficulty of integrations.

A Simple Method for Constructing a Differential Curve (*Ein einfaches Verfahren zur Bildung von Differentialkurven*, R. Slaby. *Zeits. des Vereines deutscher Ingenieure*, vol. 57, no. 21, p. 821. 2 pp. 3 figs. *p*). It is often necessary to construct a curve that would show the differential of another curve, e.g., a velocity or acceleration curve from a time-path curve. The author recommends the following simple method of construct-

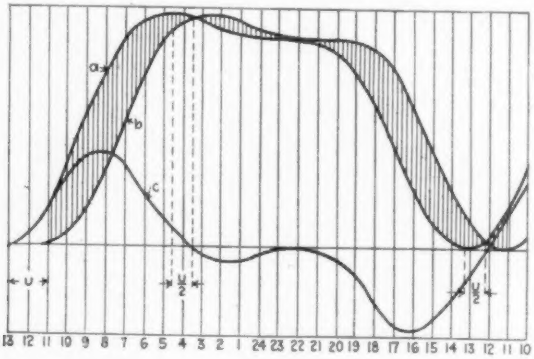


Fig. 6 Differential Curve Construction

ing such curves. The original curve a (Fig. 6) to which a differential curve has to be constructed, is displaced, parallel to the axis of abscissae, through some small distance u, and the difference in ordinates between the original and derived curve b plotted as a curve: *this curve is the desired differential curve.* Since, however, it is displaced in position from the original curve, it may be shifted through $u/2$ towards the origin, in order to bring it into complete correspondence with the curve a. The author proceeds to prove mathematically that the curve of differentials so obtained is not an approximation, but practically an exact curve of differentials. This part of the article is omitted owing to lack of space.

Internal Friction of Rarefied Gases (*Über die innere Reibung verdünnter Gase*, A. Timiriazeff. *Annalen der Physik*, ser. 4, vol. 40, no. 5, p. 971, 1913. 20 pp., 10 figs. *etA*). Only the conclusions of this interesting article can be given here. The author describes a process for the determination of internal friction in rarefied gases. In the theoretical part of his investigation he follows mainly the Maxwell-Bolzmann method and determines the variation of quantity of motion as a function of pressure. He finds that: (a) as has been established by previous experimenters, the sliding of gas particles along the surface of a solid body in contact with the gas is inversely proportional to the density of the gas,

and therefore can be observed only in the case of gases of comparatively low density; in this case, however, the magnitude of the sliding is $a_o/p = c_o\lambda$, and is therefore directly proportional to λ, average free length of the path, and inversely proportional to the pressure p. (*b*) The coefficient of sliding a_o is related to the coefficient of sudden change of temperature γp (introduced by Smoluchowski) in accordance with the following simple equation: $a_o = \frac{8}{15}\gamma p$. (*c*) The coefficient of sliding a_o has been determined by measuring the sudden changes of temperature, and the theoretical curve G of variation of quantity of motion was

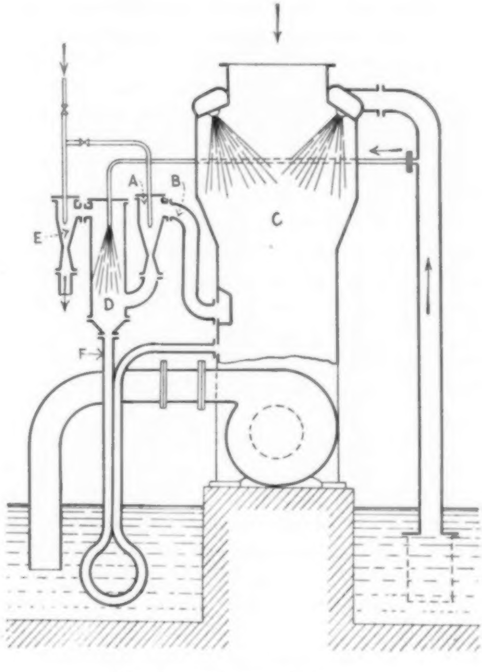


FIG. 7 BREGUET EJECTAIR

plotted as a function of log p. Experiments with air and carbon dioxide have confirmed the correctness of the above. (*d*) The curve $G = f$ (log p) has an inflection point corresponding to pressure p_1 inversely proportional to the thickness of the gas layer d. The above does not apply to cases of very high rarefication.

Steam Engineering

BREGUET EJECTAIR (*L'Ejectair Breguet*, H. Eric. *Revue industrielle*, vol. 44, no. 2085/20, p. 265, May 17, 1913. 2 pp., 4 figs, *d*). Description of a new apparatus for producing vacuum in condensers of steam engines. Neither the principle of the apparatus nor its construction are really new, but the designer of the present device, Maurice Delaporte, of the Breguet Works, of Paris, France, was the first to construct one that would work

economically. The distinctive feature of the Breguet Ejectair is the use of two ejectors in series (Fig. 7). The first one *A* takes steam, by the pipe *B* from the condenser *C* in which a vacuum has to be produced, and drives the mixture of air and steam into an auxiliary condenser *D*, where the air is picked up by the second steam ejector *E*, while the water collecting at the bottom of the auxiliary condenser *D* is directed, by the pipe *F* (notice its shape) to the water pump of the main condenser. The pipe *F* is shaped as shown in order to make it answer its three purposes: to permit maintenance of a difference of pressure between the two condensers, allow water to flow, and prevent the flow of air. The only pump with moving parts in this apparatus therefore takes care only of water, and therefore an ordinary centrifugal pump of simple and rugged construction may be used. The name ejectair has been given to the combination of two ejectors in series and auxiliary condenser. This device was further improved by providing for an injection of cold water into the pipe *B* by which the ejector *A* takes the gaseous mixture from the main condenser. By this means part of the vapor in the mixture is condensed, and its air contents increased, so that, for the same volume handled by the ejectair, the actual volume of air extracted from the condenser becomes considerably higher. The Breguet apparatus can be easily adapted for use with surface condensers. All the steam used by the ejectors is exhausted into the feedwater tank; the apparatus is started simply by admitting steam to the ejectors, no priming being necessary; practically no attendance is required since even considerable variations of pressure do not affect the vacuum produced. (From paper read before the French Society of Civil Engineers.)

INFLUENCE OF AIR SUPPLY ON SMOKING FIRES (*Einfluss der Luftzuführung bei qualmenden Feuern*, de Grahl. *Zeitschrift für Dampfkessel und Maschinenbetrieb*, vol. 36, no. 21, p. 251, May 23, 1913. 2½ pp., 2 figs. *ep*). The author describes tests made to determine whether the cutting out of smoke consuming apparatus on locomotives and trailers affects the steam production and efficiency of the boiler, and if so, how. A Marcotty smoke consumer was used, the gases being collected shortly after new fuel had been thrown in in glass balloons, and subsequently tested in the laboratory. This proved to be more convenient than the usual tests on the spot with an Orsat apparatus inasmuch as it permitted the determination, in addition to (CO_2) and (O_2), of the contents of carbon monoxide, hydrogen and methane (CH_4). The author points out that in many tests the efficiency of a boiler plant is determined from the steam making capacities, without considering that, with a high level of water in the boiler, some of the water is carried away by the steam, and the result is apt to become misleading. Unless the water level is kept low, the results as to fuel utilization may be quite wrong, and differ materially from values obtained with a different water level. In addition to the usual determination in tests of efficiency of smoke consuming apparatus, the unburned gas particle should be determined as they affect the heat losses. Table 1 shows the difference of operation with the smoke consumer in action and cut off, and proves the importance of heat losses due to escape of CO, H_2 and CH_4 through smoke emission of the fire after fresh stoking, losses which

are to a large extent eliminated by the application of the smoke consuming device, i.e., use of overgrate blast. Attention is also called to the temperatures of flue gases, from which very misleading conclusions could easily be drawn. In fact, in test no. 1, with the smoke consumer cut out, the flue gas temperature is 305 deg. cent. (581 deg. fahr.), with the Marcotty apparatus in action 420 deg. cent. (788 deg. fahr.): the lower temperature in the first case is, however, due to incomplete combustion, and canot be used as an argument for the presence of higher efficiency.

On the whole the author found that, with the smoke consuming device cut out, the smoke stack losses were 10.41 per cent, and the losses through incomplete combustion (no attention was paid to carbon particles in the flue gases) 18.71 per cent; with the Marcotty device in operation, the respective losses were 19.85 and 0.47 per cent, thus giving a total distinctly

TABLE 1 SMOKE CONSUMER TESTS

Time	CO_2	O_2	CO	H_2	CH_4	N_2
Marcotty Smoke Consumer Cut Out						
1.26	13.65	2.00	2.81	1.31	0.48	79.75
1.26¾	13.20	1.35	5.13	1.90	0.57	77.85
1.27½	13.65	1.35	3.68	1.61	0.40	79.31
1.28¼	14.60	2.20	2.77	0.46	0.22	79.75
Marcotty Smoke Consumer in Operation						
12.25½	12.20	5.50	0.22	0.07	0.11	81.90
12.26¼	12.70	6.00	....	....	....	81.30
12.27	15.00	4.00	....	....	....	81.00
12.27¾	13.00	6.50	....	....	....	80.50
12.29	12.00	7.50	....	....	....	80.50

in favor of the smoke consumer operation. As the period after stoking increases, the two sets of figures naturally approach one another. The article contains a description of the author's tests which cannot be more fully reported here owing to lack of space.

This table shows that while the smokestack losses are larger when the smoke consumer is used, the losses through incomplete combustion of gases are in that case so small as to make the final result materially in favor of the use of the smoke consumer. This is further illustrated in the original article by diagrams. The author states that firemen are usually against using visible air admission, because it may be easily abused and thus produce fall of vacuum, decrease of steam production, and increased fuel consumption. It is therefore advisable to install, in connection with smoke consuming apparatus, devices for automatically ("invisibly to the fireman," as the author expresses it) regulating the air admission, and

at the same provide means for restoring the vacuum produced by supplementary air admission. In locomotives an auxiliary air blower should be provided which would automatically begin to act as soon as the steam was shut off, so as to provide the air necessary for a proper consumption of coal. All the devices concerned with smokeless coal burning on the locomotive must be automatic, since the fireman's work is so strenuous as to leave him no time for their proper handling.

THE MODERN PROBLEMS OF THE FURNACE ROOM AND THEIR SOLUTION (*Sovremeniya zadachi kachegarki i ikh reshenye*, K. I. Plamenevski. *Zapiski of the Russian Imperial Technical Society* (in Russian), vol. 47, no. 3, p. 67, March 1913, serial, not finished. *cd*). General discussion and comparative description of various types of *boiler furnaces*, among others

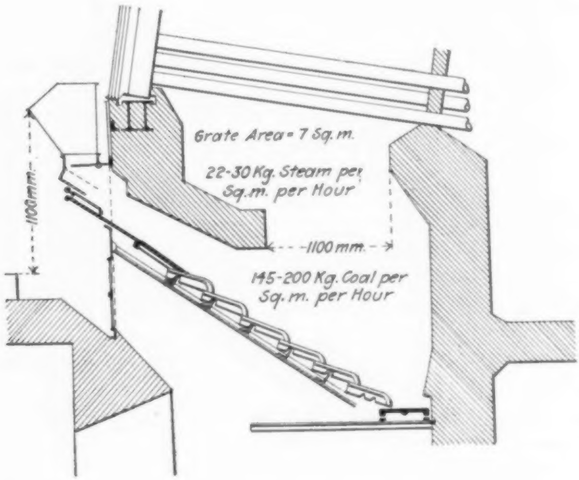


FIG. 8 LOMSHAKOFF FURNACE

that of the *Lomshakoff furnace*, tested at the Nevsky shipbuilding works in St. Petersburg, Russia. It is of the inclined grate type (Fig. 8), with arches directing the flow of gases over the incandescent fuel, these gases being conducted to the smokestack through a narrow pass which was expected to assist in complete mixing and combustion of gases. At first the arch was made very long, and the pass narrow, on the supposition that the narrower the pass, the more complete will be the combustion of the gases. It was found however that the distribution of temperature throughout the boiler was very unequal, and at the pass into the smokestack so high that firebricks melted and the arches fell away in less than one month. Long grates were used, with the result that coal, ashes, slag and bricks accumulated under the bars, and not less than four men had to be employed to clear it away. The long and heavy grate bars easily burned off, and as soon as about 4 in. of their length was destroyed, the bar had to be taken out, with a great loss of time and effort. Numerous tests were

made and finally the design shown in Fig. 8 was evolved, with a short arch (400 mm, or 15.7 in.), large gas combustion space, short grate bars disposed in seven steps as shown, and a more convenient system for the removal of slags. Thus reconstructed, the furnace has been in operation for over a year without repairs, burning 180 kg of coal per qm per hour (36.8 lb. per sq. ft. per hr.), with temperature at the furnace bridge 250 to 300 deg. cent. (662 to 572 deg. fahr.), and draft 6 to 10 mm (0.23 to 0.39 in.) with half closed damper, while with 20 mm (0.78 in.) draft 200 kg per qm (41 lb. per sq. ft. of coal can be burned, and 30 kg of steam per qm (6.15 lb. per sq. ft.) of heating area developed. The normal content of CO_2 from 10 to 13 per cent; CO none. At the same place tests have been made with compressed air undergrate blowing in connection with an ordinary grate, and it was found that the combustion of fuel proceeds more economically and can be more completely regulated than otherwise. With air pressure in the undergrate blower of 25-30 mm (1 to 1.18 in.) it was easy to burn 140 kg per qm (28.8 lb. per sq. ft.) grate area. The difficulty with which they had to contend in these tests was due to the absence of apparatus showing the efficiency of furnaces with undergrate draft, CO_2 recording in itself not being sufficient. These tests have, however, indicated the general usefulness of undergrate blowing with compressed air, as well as the necessity for a different type of grate bar, such as would eliminate the loss of compressed air; some method of obtaining a uniform distribution of the air blast through the fuel bed is also wanted.

Modern Experiments and Experiences with Materials for Turbine Blades for High Temperatures (*Neuere Versuche und Erfahrungen mit Turbinen-schaufelmaterial für hohe Temperaturen*, Schulz. *Die Turbine*, vol. 9, nos. 13, 14, 15, pp. 225, 243, 266, April 5 and 20, and May 5, 1913. 7 pp. *ep*). A complete compilation of data on materials for use in *superheated steam and gas turbine* construction for parts exposed to the action of hot gases and at the same time to high stresses. The author takes up one material after another, citing data of tests on its tensile strength and elongation at temperatures up to or above 500 deg. cent. (932 deg. fahr.), giving references to former investigations. Notwithstanding its high melting point, pure nickel cannot be used on account of its fragility at high temperatures. Wrought iron appears to be very suitable for superheated steam turbines [tensile strength at 932 deg. fahr., according to Dr. Kollmann, fell from 37 kg (52,600 lb. per sq. in.) to 13 kg (18,400 lb. per sq. in.), but according to Rudeloff at 400 deg. cent. (752 deg. fahr.) it still has a tensile strength of 32 kg (45,500 lb. per sq. in.) and an elongation of 40 per cent]. It is used for blades in the Thyssen and Melms & Pfenninger steam turbines. Ordinary bronzes have not proved satisfactory, but some of the special bronzes gave good results; thus, Professor Striebeck obtained good results with Durana metal (59 Cu, 40 Zn, 1 Sn, 0.4 P, 0.3 Fe) at temperatures up to 350 deg. cent. (662 deg. fahr.); nearly similar results were obtained with Resistin (a bronze containing 5 to 6 per cent of manganese): the first of these bronzes is used in the Imle turbines, the second in the blades in the superheated steam stages of the turbines of the steamer Kaiser, built by the Vulkan yards of Stettin. The Stones manganese bronze is used for blade construction in the Japanese navy. Very good results have been obtained in 1912 at the Weser Company yards in tests with Ruebel bronze, but these data cannot yet be considered as conclusive. Aluminum bronzes do not appear to have withstood long

usage, though satisfactory on short tests. As regards special steels (alloy steels), the tests of Fabry, at the laboratory of the Hungarian Royal Iron and Steel Works, have shown that at high temperatures low carbon steel is superior to high carbon. Rudeloff has found that nickel steels with 8 to 16 per cent nickel materially increase in strength when slowly cooled after heating. Carbon-free nickel-iron alloys patented for use in turbine blades by the Brown-Boveri Co., have failed in laboratory tests. Regular 5 per cent nickel steel has proved to be very efficient, while higher nickel steels (25 per cent) have not been found to be quite as good. On the other hand Guillet, on the basis of extensive tests, recommends for blades steel of 30 to 32 per cent Ni and maximum 0.12 per cent C. This steel has the further advantages that its coefficient of expansion is small, and that it does not rust in clean water. As regards Monel metal, there is little doubt as to its good qualities, especially in the composition 68 per cent Ni, 27 per cent Cu, and the rest iron and manganese, but it is so hard that it cannot be conveniently rolled, at least that has been the experience of the firm Heckmann in Germany which finally discontinued its manufacture. It makes, however, turbine blades of a softer material, with 65 per cent Ni content. Hörenz and Imle are now making tests with Monel blades in turbo-blower blades, but the data have not been published yet; it appears, however, that at red heat the metal is malleable, and after being once heated to redness does not oxidize further. The article gives also some data as to another nickel alloy called "Chronin," but does not indicate its composition. Chrome-nickel alloys, with 25 and more per cent Cr, have a high resistance to chemical influences, but are not malleable and cannot generally be shaped mechanically. The addition of 1.5 to 2.5 per cent silver (30 per cent chrome, rest Ni: patent Borchers) is said to improve materially the mechanical qualities of the alloy, so that it can be machined in the usual manner.

Strength of Materials and Materials of Construction

ON THE ELASTIC LIMIT OF ALLOYS (*Sur la limite élastique des alliages*, A. Portevin. *Comptes rendus de l'Académie des Sciences*, vol. 156, no. 16, p. 1237, April 21, 1913. 4 pp., 6 figs. *e*). One of the methods of determining the *elastic limit* of metals and alloys is that of *slip-bands*. The author made a series of tests with alloys, using, for simplicity's sake, alloys which have preserved at the ordinary temperature their crystalline structure of solidification, that is, have not after solidification passed through either intentional deformation or secondary recrystallization. The tests were made with: (*a*) alloys formed of a single, solid, chemically homogeneous, solution (slip bands appeared first in certain grains only and gradually spread out to all the grains; the sections of maximum and minimum limits of elasticity may be determined, by noting the sections where the initial and complete deformation occur first); the author observes in this connection that the limit of elasticity which is a vectorial quantity in a single grain, becomes a scalar for the entirety of the piece only because of the lack of uniformity in the orientation of the crystals. (*b*) In alloys formed of a single, solid, chemically heterogeneous, solution, the chemical composition varies in each crystalline aggregate from the center to the periphery, the central parts having the highest melting point. Each crystal will behave in the same way as in the preceding case, with that difference, however, that the elastic limit of the central part is not the same as that of the periphery, the slip-bands appearing first in the part of the crystal having the lowest elastic limit. (*c*) In the case of a complex of two phases, e. g. brass with a 57 per cent content of

copper, there are two constituents, and the slip-bands appear first in one of them (a). All these observations tend to show that the curve of load-deformation is *continuous*, and cannot be therefore set precisely in one definite quantity (elastic limit): in fact, it extends between limits which depend both on mechanical anisotropy of the crystals, and degree of chemical homogeneity of the alloy.

Lineal Expansion of Solids by Heat at Higher Temperatures (*Thermische Ausdehnung fester Körper bei höheren Temperaturen*, A. Werner. *Zeits. für Dampfkessel und Maschinenbetrieb*, vol. 36, no. 19, p. 227, May 9, 1913. 3 pp., 8 figs. *e*). Description of some experiments on, and methods used for, the

TABLE 2 LINEAL EXPANSION BY HEAT OF NICKEL STEEL

Temperature, Deg. Cent.	Lineal Expansion in millimeters per Meter of Material Percentage of Nickel in Steel			
	5	25	25	33
0	0.000	0.000	0.000	0.000
50	0.472	0.731	0.741	−0.009
100	1.003	1.564	1.608	+0.009
150	1.589	2.471	2.556	0.119
200	2.203	3.418	3.545	0.386
250	2.833	4.378	4.527	0.876
300	3.465	5.322	5.464	1.651

The two columns for 25 per cent nickel steel correspond to test pieces of different origin.

determination of the coefficient of lineal expansion of solids at higher temperature in the laboratory of the Physikalisch-Technische Reichanstalt (for a fuller description of this new apparatus see A. Leman and A. Werner, *Zeits. für Instrumentenkunde*, 1913, no. 3, p. 65). The lineal expansion is determined relatively to that of quartz glass which is convenient since the expansion of quartz glass has been determined with great precision up to 1000 deg. cent. (1832 deg. fahr.). Table 2 shows the data obtained for the expansion of nickel steel with various percentages of nickel, interesting because of the wide use of nickel steel in automobile and steam turbine construction.

Enemies of Reinforced Concrete (*I nemici del cemento armato*, Professor Rohland. *Verkehrtechnische Woche*, April 26, 1913, through *L'Ingegneria Ferroviaria*, vol. 10, no. 9, p. 137, May 15, 1913. *p*). A review of various influences affecting the strength and durability of reinforced concrete (action of water of various composition, earth, and electric currents), and explanation of the underlying chemical phenomena.

Experiments on the Distribution of Stresses in Notched Bars under Tension (*Versuche über die Spannungsverteilung in gekerbten Zugstäben*, Dr. Ing. E. Preuss. *Mitteilungen über Forschungsarbeiten auf dem Gebiete des Ingenieurwesens*, no. 134, 1913. p. 47, 15 pp., 30 figs. *e*). Experimental investigation on the distribution of stresses in notched bars under tension, to be read in connection with the author's former investigation on the distribution of stresses in punched bars (*The Journal*, February 1913, p. 341). For notched bars the author shows that (*a*) in bars with equal depth of notch, the stress at the notch edge is inversely proportional to the diameter of the notch; (*b*) in bars with equal

diameters of notch and equal notch widths, the stress at the notch edge is proportional to the depth of notch; (*c*) when the notch is semicircular, the stress at the notch edge is inversely proportional to the diameter; (*d*) it has been found that, with the exception of the case of sharp cornered notches, the stress at the notch edge is from 1.43 to 2.48 greater than the usually assumed average stress; (*e*) the minimum stress along the middle axis of the bars in the bars tested is subject to stresses 0.71 to 0.98 times the average stresses.

Miscellanea

Washing, Bathing and Dressing Rooms in Factories (*Wasch- und Bade-, sowie Ankleideraume in Fabriken*, H. W. *Sozial-Technik*, vol. 12, no. 10, p. 187, May 15, 1913. 3 pp. *p*). A practical discussion of the question of providing workmen with washing, bathing and dressing facilities, and the best types of furniture and apparatus to be used in this connection. The author points out that the fact that the workmen sometimes do not make as much use of the facilities provided for them as could be expected, is due mainly to the unsatisfactory arrangement or location of the rooms, forcing the men to lose much time in reaching them or in waiting for their turn to use the washstands. Often also the washrooms are kept in a state such as to make their use repulsive. On the other hand, the training of workmen in orderliness and cleanliness is of advantage both to the men and the employer, and in some trades, e. g. where poisonous substances are used, of absolute necessity. The author recommends the use of separate washrooms for the different shops rather than a common place for the entire factory, both because of their better accessibility, and because it is advantageous to keep the separate groups of men as much apart as possible.

Smokestack Construction with Special Regard to Accident Prevention (*Schornsteinbau unter besonderer Berücksichtigung der Unfallverhütung*, H. Dieckhoff. *Sozial-Technik*, vol. 12, no. 9, p. 161, May 1, 1913. 8 pp., 7 figs. *dA*). Discussion of the main *safety devices* to be used in smokestack construction. The author is owner of a well-known firm of smokestack constructors in Germany. The removal of old smokestacks is also briefly discussed.

Sound Conductivity of Building Materials and Walls (*Über Schalldurchlässigkeit von Baumaterialien und ausgeführten Wänden*, R. Ottenstein. *Gesundheits-Ingenieur*, vol. 36, no. 19, p. 345, May 10, 1913. 4 pp., 3 figs. *e*). A preliminary publication of data of an investigation on *sound conductivity* in various materials and structures. Information about this investigation has already been published in *The Journal* (March 1912, p. 430); a fuller account will be given after the complete publication of the data of the original investigation. From the data published in this article it appears that the increase in the weight of the sound-protecting plate or wall is proportional to the useful effect obtained only up to a certain limit, beyond which the sound conductivity of the plate decreases much slower than the increase in weight. The fundamental condition of sound protection is the best possible airtight division between the spaces in question; heavy walls afford a better protection against transmission of sound, due to their higher resistance to the rise of oscillations; air spaces between oscillating walls lessen the dampening action of the walls; air spaces between porous walls or a porous and a solid wall are harmful, and packing of loose material between such walls is of little help, since it does not offer sufficient resistance to the propagation of sounds. A method for the determination of sound conductivity in walls is described.

ARTICLES UPON GAS POWER

(Prepared by the Gas Power Literature Committee)[1]

BETRIEBSERFAHRUNGEN MIT DIESELSCHIFFEN. *Zeitschrift des Vereines deutscher Ingenieure*, March 29, 1913. ½ p. *p*.

Marine operating experience with the Diesel engine.

FLUGZEUGMOTOR, DIE DURCHFUHRUNG UND DAS ERGEBNIS DES WETTBEWERBES UM DEN KAISERPREIS FÜR DEN BESTEN DEUTSCHEN, F. Bendemann and Seppeler. *Zeitschrift des Vereines deutscher Ingenieure*, May 3, 1913. 6 pp., 23 figs. *p*.

The conduct and result of the competition for the Emperor's prize for the best German flying-machine motor.

GEMISCHBILDUNG IN GASMASCHINEN, ZEICHNERISCHE UNTERSUCHUNG DER, J. Magg. *Zeitschrift des Vereines deutscher Ingenieure*, May 3, 1913. 3½ pp., 3 curves. *m*.

Graphic investigation on the mixture formation in gas engines.

STEUERUNGSDIAGRAMM FÜR VIERTAKTMASCHINEN, J. Magg. *Zeitschrift des Vereines deutscher Ingenieure*, February 15, 1913. 2 pp., 6 figs. *m*.

Article on valve-gear diagram for four-stroke cycle engines.

VERBRENNUNGSKRAFTMASCHINEN UNS DER WELTAUSSTELLUNG IN GENT, P. Meyer. *Zeitschrift des Vereines deutscher Ingenieure*, May 17, 1913. Serial article. *dp*.

Internal-combustion engines at the World's Fair in Ghent.

VERBRENNUNGSMOTOREN UND EIN NEUER SECHSTAKTMOTOR, DIE STEIGERUNG DER LEISTUNG VON, Emil Schimanek. *Zeitschrift des Vereines deutscher Ingenieure*, January 25, 1913. 7¾ pp., 11 figs., 2 tables, 8 curves. *mp*.

Increasing the duty capacity of internal-combustion engines and a new six-stroke cycle engine.

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[1] Opinions expressed are those of the reviewer, not of the Society. Articles are classified as *c* comparative; *d* descriptive; *e* experimental; *h* historical; *m* mathematical; *p* practical. A rating is occasionally given by the reviewer, as *A*, *B*, *C*. The first installment was given in The Journal for May 1910.

MEETING

SAN FRANCISCO MEETING, MAY 15

A meeting of the San Francisco Section of the Society was held on Thursday evening, May 15, in the rooms of the Commercial Club. G. L. Bayley, Mem.Am.Soc.M.E., chief mechanical and electrical engineer of the Panama-Pacific International Exposition Company, read a paper on the progress of the buildings being erected for the exposition, illustrating his remarks with lantern slides.

BOSTON MEETING, MAY 23

A meeting of the Boston Section of the American Institute of Electrical Engineers, in which the members of the Society were invited to participate, was held on Friday evening, May 23, in the Edison Company Auditorium. The paper of the evening was upon the Organization and Methods of a Large Engineering and Construction Company, presented by Roy M. Henderson, vice-president and construction manager of the Stone & Webster Engineering Corporation. In it he sketched the steps in the development of the business leading up to the present organization and outlined the latter in detail with special reference to the engineering, construction, purchasing and accounting staffs. He described also the scheme of local organization for field operations.

STUDENT BRANCHES

CASE SCHOOL OF APPLIED SCIENCE

On May 8, the Mechanical Engineering Club of Case School of Applied Science elected the following officers for the coming year: chairman, H. C. Mummert; vice-chairman, S. Kenyon; secretary, C. Stemm; treasurer, S. Stanley; senator, L. F. Milligan.

The paper of the evening was Coal and Ashes Handling Machinery, by David Gaehr, Mem.Am.Soc.M.E. It was the first time the lecture was given and the author had prepared many original slides to illustrate his talk. A social gathering followed.

PURDUE UNIVERSITY

At a meeting of the Purdue Student Section held May 20, the following officers were elected for the coming year: chairman, A. D. Meals; vice-chairman, J. M. Lonn; recording secretary, R. E. Kriegbaum; treasurer, W. T. Miller; program committee, S. A. Peck and F. G. Spencer; corresponding secretary, G. F. Lynde; members of the governing council, C. W. Handley and E. A. Tuttle.

UNIVERSITY OF CINCINNATI

The annual election of officers of the Student Branch of the University of Cincinnati was held May 27 with the following results: chairman, A. O. Hurxthal; vice-chairman, R. M. Race; secretary and treasurer, E. A. Oster.

Preceding the election, Charles S. Gingrich, Mem.Am.Soc.M.E., addressed the section on The Engineer, discussing the municipal and economic problems confronting him, and his relation to capital and labor.

UNIVERSITY OF ILLINOIS

At the last meeting of the year held on May 23, the Student Branch of the University of Illinois elected the following officers: chairman, A. H. Aagaard; vice-chairman, Geo. Meyer; treasurer, H. C. Peterson; secretary, H. E. Austin.

The program for the afternoon consisted of a paper on The Design of Gas Tractors, by E. J. McCormick.

NECROLOGY

ADOLPHUS BONZANO

Adolphus Bonzano, pioneer bridge builder and inventor of the Bonzano rail joint and other railroad appliances, was born at Ehingen, Germany, December 5, 1830. He received a classical and engineering education both at Ehingen and at Stuttgart, and in 1850 came to America to perfect himself in the study of English. In 1851 he went to Springfield, Mass., where for the following four years he served as apprentice, machinist and draftsman for the American Machine Works. Until 1860 he was engaged as superintendent of construction of the Detroit Dry Dock Iron Works, which was later transformed into the Detroit Bridge & Iron Works, one of the earliest bridge building plants in this country. In 1868 he moved to Phoenixville, Pa., where with Thomas Curtis Clarke and others he formed the firm of Kellogg, Clarke & Company, bridge builders, he acting as chief engineer. In 1884 this firm was dissolved and was succeeded by the Phoenix Bridge Company, of which Mr. Bonzano was made vice-president and chief engineer. In 1893 he resigned this position and with Mr. Clarke opened an office in New York as consulting engineers. After his partner's death in 1898, Mr. Bonzano retired from active business to devote himself to the invention of railroad and other appliances. The Pecos viaduct on the Southern Pacific Railroad in Texas, the Kinzua viaduct on the Erie Railroad, and the Chesapeake & Ohio bridge at Cincinnati are among the more notable examples of his work. He died May 5, 1913.

HORATIO A. FOSTER

Horatio A. Foster was born at Bustleton, Philadelphia, Pa., January 12, 1858. His engineering training began in the fall of 1884 with the Daft Electric Company, Greenville, N. J.; the next year he went to Baltimore to electrify a short branch of the Baltimore Union Passenger Railway Company. In 1886 he entered the shops of the Thomson-Houston Electric Company at Lynn, Mass., and in September 1888 was appointed superintend-

ent of the East River Electric Light Company, New York, remaining with that company till July 1891. He was then appointed an expert for the United States Census office to compile data on the electrical industry of New York State. In May 1893 he accepted a position in the editorial department of Electric Power, and later in the same year became associated with George Forbes, electrical engineer of the Niagara Falls Power Company, and had charge of his New York office for about a year and a half. In 1895 Mr. Foster joined the staff of the Cataract Construction Company of Niagara Falls as testing engineer. After several years in general consulting work he became interested in the valuation of public utilities, studying traffic conditions and other matters pertaining to public service, being engaged in this work with J. G. White & Company at the time of his death, April 27, 1913.

Mr. Foster was the author of the Electrical Engineers Pocket-Book, which bears his name, also Valuation of Public Utilities, and he had frequently contributed to the technical press. He was a member of the American Institute of Electrical Engineers, the Engineers Club of New York, and the Philadelphia Arts Club.

PETER KIRKEVAAG

Peter Kirkevaag was born in Christianssund, Norway, April 1849, and in 1871, after finishing an apprenticeship, he went to Germany with a stipend from the Norwegian government. He was graduated from the polytechnic school in Langensalza, Turingen, in 1874 and was afterwards employed as draftsman and engineer in Westphalia until 1877, following which he was inspector for three years for the Nordenfelth gun factory in Stockholm, Sweden. In 1881 he came to the United States, where he secured employment as machinist and draftsman with Oliver Brothers & Phillips and A. Garvison & Co., Pittsburgh, Pa. Two years later he became draftsman and superintendent of buildings, foundations, etc., for the Hartman Steel Company, Beaver Falls, Pa., and this same year saw the beginning of his connection with William Tod & Company, Youngstown, Ohio. Thirty years later he left this company to become associated with the Brier Hill Steel Company of the same town. He died May 6, 1913.

HAROLD SERRELL

Harold Serrell was born in Brooklyn, N. Y., August 26, 1852. Having completed his education at Adelphi Academy in 1869,

he entered the office of his father, Lemuel Wright Serrell, patent attorney and expert in patent causes, with whom he was associated under the firm name of L. W. Serrell & Son. After his father's death in 1899, he carried on the business alone, studying and familiarizing himself with machinery, mechanical devices and the arts and sciences for the professional career of solicitor of patents and mechanical expert. He served in contested cases as mechanical expert, giving testimony both in court and before a master for use in court. He died February 26, 1913.

OLIN SCOTT

Olin Scott was born February 27, 1832, at Bennington, Vt., where he learned the trade of a millwright. In 1858 he formed a partnership with H. S. Brown and established the Bennington Machine Works; in 1863 he purchased the interests of his partner and in 1864 purchased and combined the business of the Eagle Foundry and Machine Works. Later this foundry developed into a factory for the manufacture of powder mill and pulp mill machinery. Before the use of nitro powders became general, the Bennington Machine Works had acquired a national importance and worldwide reputation, machines from its shops having been shipped to every continent on the globe. After the close of the Civil War Colonel Scott ceased to confine his energies exclusively to manufacturing and became an organizer of powder manufacturing companies. In 1869 he built the Lake Superior Powder Mills at Marquette, Wis., and four years later he became superintendent of the Laflin and Rand Powder Company of New York. In 1882 he organized the Ohio Powder Company of Youngstown, Ohio, and for several years was vice-president of the corporation. He also organized the Pennsylvania Powder Company at Scranton, Pa., in 1884, becoming its president. Three years later he disposed of his interests in Ohio and Pennsylvania and became a consulting engineer for the Laflin and Rand Company and the Dupont Powder Company of Wilmington, Del., a position he retained until 1894. In 1892 he became president of the Lasher Stocking Company and operated the property until its comparatively recent disposition to the Vermont Hosiery and Machinery Company.

Colonel Scott died in Bennington, April 28, 1913.

ACCESSIONS TO THE LIBRARY

With Comments by the Librarian

This list includes only accessions to the library of this Society. Lists of accessions to the libraries of the A. I. E. E. and A. I. M. E. can be secured on request from Calvin W. Rice, Secretary, Am. Soc. M. E.

Academy Architecture and Architectural Review. Classified index to vols. 1–21, 1889–1902. *London, 1902.*

American Wood Preservers' Association. Proc. 9th annual meeting, 1913. *Baltimore, 1913.* Gift of association.

Asphalt Construction for Pavements and Highways, Clifford Richardson, 1913.

Association of Railway Electrical Engineers. Proc. vol. 5, 1912. *Chicago, 1912.* Gift of association.

British Fire Prevention Committee. Journal no. 10, 1913. *London, 1913.*

Cassier's Magazine (London). Oil Power Number, March 1913. *London, 1913.* Gift of W. R. Haynie.

Celluloid Dangers with Some Suggestions. British Fire Prevention Committee, no. 179. *London, 1913.*

Connecticut. Bureau of Labor Statistics. 25th Report, 1912. *Hartford, 1912.* Gift of bureau.

Conservation of Water Powers, R. G. Brown. *Washington, 1913.*

Dictionary of Automobile Terms, A. L. Clough. *New York, 1913.*

Diesel Liquid Fuel Engine, W. J. Dyer. Before Hawaiian Engineering Association. *Honolulu, 1913.* Gift of author.

The Engineer's Yearbook for 1913, H. R. Kempe. *London, 1913.*

Gas Engine in Modern Blast Furnace and Steel Plants, Heinrich J. Freyn. Gift of author.

Gasoline Automobile, Its Design and Construction, P. M. Heldt. vol. 2. Transmission, running gear and control. *New York, 1913.*

Industrie und Ingenieurwerke in Mittel und Niederschlesien. Festschrift zur 52 Hauptversammlung des Vereines deutscher Ingenieure in Breslau June 10–14, 1911. *Breslau, 1911.* Gift of Verein deutscher Ingenieure.

Investigation of the conditions governing thr choice of a Proper Quality Standard for artificial gas with conclusion and recommendation of the Joint Committee on Calorimetry of the Public Service Commission and Gas Corporations in the Second Public Service District New York State. Gift of Empire State Gas and Electric Association.

Leland Stanford Junior University. Register 1912–1913. Gift of university.

Light: Its Use and Misuse. A primer of illumination prepared under the

direction of the Illuminating Engineering Society. *New York, 1912.* Gift of Calvin W. Rice.

Lowell Textile School. Bull. 1913–1914. *Lowell, 1913.* Gift of school.

Die Maschinen-Getriebe, Wilh. Hartmann. vol. 1. *Stuttgart, 1913.*

Mecanique, Electricité et Construction Appliquées aux Appareils de Levage, Louis Rousselet. vols. 1 and 2. *Paris, 1908, 1913.* Gift of Hunt Memorial Fund.

Milwaukee, Wis. Smoke Inspector. Annual Report 1912. *1912.*

Monthly Official Railway List, May 1913. *Chicago, 1913.*

New York State Charities Aid Association. Annual Report 1912. *New York City, 1912.* Gift of association.

Ohio University. Catalogue 1913–1914. *Athens, 1913.* Gift of university.

Principles of Irrigation Engineering, F. H. Newell and D. W. Murphy. *New York, 1913.*

Purification of New York Harbor, G. A. Soper. Reprinted from the Medical Record, May 17, 1913. *New York.* Gift of author.

Rhode Island. Public Utilities Commission. Annual Report 1912. *Providence, 1913.* Gift of commission.

Schweizerischer Ingenieur und Architektenverein. Staats und Handelswissenschaftlicher Kurs, 1913. Gift of Schweizerischer Ingenieur und Architektenverein.

Specifications for Street Roadway Pavements, S. Whinery. ed. 2. *New York, 1913.*

Stone & Webster Engineering Corporation. Brief data regarding various water power plants. *Boston.* Gift of Calvin W. Rice.

Street Railway Journal. Advertisements, vols. 7–32. *New York, 1891–1908.* Gift of H. W. Blake.

Technischen Staatslehranstalten in Chemnitz. Reiseberichte über das technische und gewerbliche schulwesen Nordamerikas. Abhandlungen und Berichte February 1913. *Chemnitz, 1913.* Gift of Technischen Staatslehranstalten in Chemnitz.

Treatise on Hydraulics, W. C. Unwin. ed. 2. *London, A. & C. Black, 1912* Gift of author.

The first edition was published in 1907. This second edition has added a short summary of more recent researches on the measurement of flow.

U. S. Interstate Commerce Commission. 26th Annual Report. *Washington, 1913.* Gift of commission.

University Business Administration, J. C. Christensen. *1912.* Gift of author.

Visit to Germany of the American Society of Mechanical Engineers on the Invitation of the Verein deutscher Ingenieure, 1913. Gift of Verein deutscher Ingenieure.

Die Wasserkräfte ihr Ausbau und ihre wirtschaftliche Ausnutzung, Adolf Ludin. vols. 1–2. *Berlin, 1913.*

Zeitschriften der gesamten Eisenbetonliteratur 1912, A. Fitzinger. *Berlin, 1913.*

EXCHANGES

Deutschen Museums. Verwaltungs-Bericht über das neunte Geschäftsjahr 1911–1912. *München, 1912.*

Liverpool Engineering Society. Trans., vol. 33. *Liverpool, 1912.*

MUNICIPAL ENGINEERS OF THE CITY OF NEW YORK. Proc. 1912. *New York, 1913.*

SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS. Trans. vol. 20, 1912. *New York, 1913.*

UNITED ENGINEERING SOCIETY

KANSAS GAS AND ELECTRIC LIGHT STREET RAILWAY AND WATER ASSOCIATION. List of Members. *1913.* Gift of association.

LIBRARY OF CONGRESS. Classification. Class Q, Science. *Washington, 1913.* Gift of library.

NEW YORK PUBLIC LIBRARY. List of Works relating to Electric Welding. *New York, 1913.* Gift of W. B. Gamble.

REPORT OF THE COMMITTEE APPOINTED PURSUANT TO HOUSE RESOLUTIONS 429 AND 504 TO INVESTIGATE THE CONCENTRATION OF CONTROL OF MONEY AND CREDIT. February 28, 1913. Gift of House.

TRADE CATALOGUES

BRISTOL COMPANY, *Waterbury, Conn.* Bull. 143, Recording differential pressure gages and recording flow-rate meters, April 1913; Bull. 170, Patent electric furnaces for soldering coppers, April 1913; Cat. 173, Recording differential pressure gages and recording flow-rate meters. Float type, model 1010, April 1913; Cat. 1300, Bristol's Class III frecording thermometers, April 1913.

CHICAGO PNEUMATIC TOOL CO., *Chicago, Ill.* Bull. 34L, General pneumatic engineering information, April 1913; Bull. 128, Miscellaneous equipment for pneumatic drills, April 1913; Bull. 132, Pneumatic motors and pneumatic geared hoists, April 1913; Bull. 138, Cylinder air hoists and jackets, April 1913.

HOLOPHANE WORKS, *Cleveland, Ohio.* Bull. 28, Decorative shades, March 1913.

JOHNS-MANVILLE CO., *Cleveland, Ohio.* J-M Power Expert, May 1913.

WESTINGHOUSE-CHURCH-KERR & CO., *New York, N. Y.* Work done (no. 5), railroad shop edition, 47 pp.

NORTH WESTERN EXPANDED METAL CO., *Chicago, Ill.* Expanded metal construction, June 1913.

EMPLOYMENT BULLETIN

The Society considers it a special obligation and pleasant duty to be the medium of securing positions for its members. The Secretary gives this his personal attention and is pleased to receive requests both for positions and for men. Notices are not repeated except upon special request. Names and records, however, are kept on the current office list three months, and if desired must be renewed at the end of such period. Copy for the Bulletin must be in hand before the 12th of the month. The published list of "men available" is made up from members of the Society. Further information will be sent upon application.

POSITIONS AVAILABLE

601 First class automobile engineer or designer for Indianapolis concern. Familiar with latest pleasure car construction. Must have had several years' experience in engineering department of established automobile concern.

603 New York concern wants draftsman for heating, ventilating and piping work. Salary \$20-\$25.

604 Head draftsman thoroughly familiar with construction of cranes and hoists, both hand and power and electric. Must be man of executive ability and capable of taking contract through.

605 Head of engineering department for concern manufacturing boilers, engines, tanks and water-heaters. Salary ranging from two to three thousand dollars a year. Location Dayton, Ohio.

606 Wanted by agricultural machinery works in Middle West, employing 700 men, rate setter qualified by thorough experience in machine shop to figure accurately allowed times on various operations, both wood and metal work. Practical man preferred. Must be broad minded and deal fairly with men, yet safeguard interests of the management. Position one of importance and responsibility.

607 Wanted a rate setter, a man who can operate all kinds of machine tools, especially turret lathes, millers, and grinders, and has had experience making time studies for piece-rate setting. Location New York State.

MEN AVAILABLE

139 Junior member, A.B. Yale, M.E. Columbia, would like to associate with engineer or firm making specialty of design and construction of industrial plants. Experienced as superintendent of construction, assistant to works manager, etc.

140 Member, Junior grade, Cornell graduate, with three years' experience in large public service corporation. Available about August first.

141 Member, technical graduate, three years' experience as instructor in mechanical engineering, eight years' experience as draftsman, head draftsman, assistant superintendent and sales engineer with responsible engine building concerns, would consider position as instructor in reputable institution.

142 Member, technical graduate, desires position as sales engineer with

high class concern. Has had wide experience as draftsman, head draftsman, assistant superintendent and sales engineer with reputable engine builders.

143 Member desires change of location. Graduate mechanical engineer with twenty years' experience in the design and manufacture of gasoline, steam, and electric locomotives, gas engines; good organizer and up-to-date in modern shop management. Prefers position as superintendent or assistant east of Mississippi River.

144 Graduate in mechanical engineering; experience in design and construction and testing of boilers.

145 Junior, technical graduate, nine years' experience in power plant and factory work, desires position as assistant to works engineer or manager; or teaching in steam and experimental engineering.

146 Mechanical engineer, 29, graduate M. I. T., excellent experience in design of industrial plants and mechanical equipment of buildings, desires permanent connection with firm of consulting or mill engineers or position as plant engineer. Location in or near New York.

147 Member, mechanical engineer and expert machinist, with ten years' experience in manual, industrial and vocational education, desires executive position along this line; best references.

148 Member desires new connection as chief engineer and factory architect or chief construction engineer. Successful designer of large manufacturing plants and special machinery, including structural steel, reinforced concrete, power plants, heating, ventilation and refrigeration. For several years head of department of mechanical engineering in large college and is open for similar position at good institution.

149 A mechanical engineer, with nine years' experience with engine works and twelve years as dean of a prominent college of mechanical and electrical engineering, has been granted a leave of absence for the coming year and would like to become associated for the next twelve or fourteen months with some reputable firm of engineers.

150 Member, graduate of Stevens Institute and post-graduate Cornell, at present dean of engineering and professor of mechanical engineering in Western college, desires to make a change.

151 Member, specialist in steam turbines, desires to make connection with firm for the manufacture of fully developed and tested cheap commercial turbine, suitable for sizes up to 1000 h.p., condensing as well as non-condensing operation.

152 Manufacturing accountant and practical shop man, 18 years' experience with best accounting and shop practice connected with steel and rolling mills, iron, steel and brass foundries, bridge and structural shops, ship yards, machine shops, woodworking shops, etc., familiar with manufacture of internal combustion engines, stationary and tractor; steam and power pumps, threshing machinery and electrical machinery, etc.; competent to handle factory reorganization work of large and small companies.

153 Mechanical engineer with several years' experience, located in manufacturing district of Eastern Pennsylvania, would like to hear from engineers or manufacturers who desire special opportunities for sales investigations without the necessity of sending own representatives.

154 Member, at present with large ball and and roller bearing concern.

as engineer in charge of automatic machinery, wishes to connect with some reliable concern manufacturing fine interchangeable parts or machines; high class executive, with 14 years' experience in the use and design of automatic machinery and standard machines of this class.

155 Mechanical engineer salesman in Cuba, Hawaii or South America; valuable experience in the producer gas power field as an inventor, designer, engineer and salesman; speaks Spanish.

156 Associate, age 35, with 17 years' broad experience in drawing-rooms on civil, structural and mechanical work, desires a position of some responsibility, in or near Philadelphia. Experience on furnaces, steel plants, mill work, power plants, chemical apparatus, gas plants, coke ovens, etc.

157 Member, at present employed, 18 years' varied experience in design and construction of machinery and buildings, remodeling, maintenance and operation of large industrial plants and equipment, systematizing of shops and processes along the lines of scientific management, testing and general plant engineering; accustomed to handling men, drawing up contracts, purchasing equipment and material, appraising properties; desires to become identified with manufacturing or industrial plant of prominence in administrative or executive position of responsibility.

158 Graduate mechanical engineer (Stevens), 23 years of age, desires position with good prospects of advancement. Has had one year's experience in full charge of general construction work, ordering all material, handling correspondence, and checking accounts, installation of safety devices. Excellent references. At present employed.

159 Business manager, 40. Twenty years' experience, comprising experimentation, designing, selling, works management, factory building and organization. Technical gradute. Has been engaged for two years in business other than manufacturing, but desires to re-enter engineering field as executive or engineering representative.

160 Mechanical engineer, technical education, 30 years of age, 10 years' practical experience, in design, construction, operation, maintenance and reorganization of mill, factory, and other manufacturing properties. Wide experience in the superintendence of central power stations, factory extension, mill and reinforced concrete construction work. Desires position of mechanical superintendent or master mechanic. Particularly experienced in practical efficiency work. At present employed.

161 Technical graduate of an Eastern school, age 26 years, would like to engage in cost and efficiency work. Experience consists of work in the shops as well as along commercial lines. Employed at present.

162 Member, mechanical engineer, desires to hear from manufacturers of power plant equipment and specialties who desire a representative in Pittsburg and vicinity. Would like to represent some home manufacturing engines, boilers, conveying apparatus or supplies; has been for fifteen years the expert in large steel manufacturing plants, listing all power machinery and acting in an advisory capacity in matters relating to improvement of power.

163 Member, 38 years of age, Cornell M.E., married. Wide experience in design, construction and erection of power plant machinery, including construction of boilers, engines, pumps, condensers, etc. At present engaged in general consulting work in power plant design and installation and mill

building, and for the past three years in general charge of testing work for one of the largest electric light power plants in the East. Would be willing to accept responsible position requiring full time.

164 Junior member, 28 years of age. Cornell M.E., four years' practical experience in design and construction of steam turbines, patent office work, street railway, etc. Desires position as assistant to executive, or as works manager or factory manager, or would accept a position which would lead to one of these.

165 Technical graduate in mechanical engineering, with master's degree, age 27, four and one-half years' practical experience, partly drafting and designing, mostly experimental work with steam and gas tractors, desires a position with a company manufacturing power farming machinery.

166 Member, age 40, holding M.M.E. degree from Cornell University, varied experience in teaching, designing, construction and operation. Especially qualified to manage an industrial plant or hold a position of economic engineer. For the past two years has managed successfully a small electrical supply house.

167 Sales manager and engineer, with 10 years' experience selling steam power plant apparatus in New York district, desires position where acquaintance among engineers and architects, heating and electrical contractors would be of advantage.

168 Graduate mechanical engineer, 5 years' experience in steam pump and air compressor work, also high speed automatic engine and boiler design, desires supervisory position with a manufacturing concern or a firm of consulting engineers. At present employed as chief engineer of a large engine and boiler shop.

169 Member, age 35, also member of two British societies, college education, medallist, 15 years' varied experience in gas, electrical, general mechanical and automobile engineering, works management, accustomed to responsible positions and control of large staff, seeks responsible post. Good references.

170 Member, 47, graduate M. I. T., experienced as sales manager and executive with large manufacturing corporations, wants position in sales or publicity department of some company convenient to New York and Connecticut. Capable of taking entire charge large selling force, organizing and conducting publicity campaigns or acting as assistant to chief executive; has made special study of scientific employment methods and individual productive efficiency. At present with well known corporation, but future possibilities too limited.

171 Member, specialist in scientific employment and personal efficiency methods, wants position with Eastern manufacturing concern having an annual pay-roll of not less than $500,000, in which definite results will assure a permanent affiliation with adequate salary.

172 Mechanical engineer with practical machine shop experience, technical education, three years' experience in elevating, conveying, mine and power transmission machinery, desires position as sales engineer with firm or representative located in Pittsburgh.

173 Young engineer, graduate Stevens Institute of Technology, experience in electrical engineering in consulting engineer's office. Desires to change for prospect of advanced work in mechanical lines.

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Terms expire 1913	Terms expire 1914	Terms expire 1915
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Stanley G. Flagg, Jr.	Henry Hess	H. M. Leland
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W. D. SARGENT (5)

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Note—Numbers in parentheses indicate number of years the member has yet to serve.

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R. C. Carpenter
Richard Hammond
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B. F. Wood
Carl Thomas
Edward Weston
W. E. Winship

Standardization of Filters

G. W. Fuller, *Chmn.*
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H. R. Cobleigh
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C. R. Birdsey
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J. E. Gibson
W. C. Kerr
T. C. McBride

New Haven

E. S. Cooley, *Chmn.*
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J. H. Lawrence
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Gas Power Committee on Meetings

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Providence Association of Mechanical Engineers

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J. A. Brooks, *Secy.*
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A. H. Whatley, *Treas.*

Note—Numbers in parentheses indicate number of years the member has yet to serve

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Institution	Date Authorized by Council	Honorary Chairman	Chairman	Corresponding Secretary
Armour Inst. of Tech.	Mar. 9, 1909	G. F. Gebhardt	H. E. Erickson	A. N. Koch
Case School of Applied Science	Feb.14, 1913	F. H. Vose	H. C. Mummert	C. Stemm
Columbia University	Nov. 9, 1909	Chas. E. Lucke	F. B. Schmidt	H. F. Allen
Cornell University	Dec. 4, 1908	R. C. Carpenter	S. D. Mills	N. C. Johnson
Lehigh University	June 2, 1911	P.B. de Schweinitz	W. C. Owen	T. G. Shaffer
Leland Stanford Jr. Univ.	Mar. 9, 1909	W. F. Durand	C. T. Keefer	K. J. Marshall
Mass. Inst. of Tech.	Nov. 9, 1909	E. F. Miller	W. H. Treat	L. L. Downing
New York University	Nov. 9, 1909	C. E. Houghton		
Ohio State University	Jan.10, 1911	Wm. T. Magruder	R. H. Neilan	R. M. Powell
Penna. State College	Nov. 9, 1909	J. P. Jackson	H. L. Swift	H. L. Hughes
Poly. Inst. of Brooklyn	Mar. 9, 1909	W. D. Ennis	B. L. Huestis	A. Bielek
Purdue University	Mar. 9, 1909	G. A. Young	A. D. Meals	G. F. Lynde
Rensselaer Poly. Inst.	Dec. 9, 1910	A. M. Greene, Jr.	E. Kneass	R. F. Fox
State Univ. of Iowa	Apr.11, 1913	R. S. Wilbur	F. H. Guldner	C. S. Thompson
State Univ. of Kentucky	Jan.10, 1911	F. P. Anderson	R. R. Taliaferro	F. J. Forsyth
Stevens Inst. of Tech.	Dec. 4, 1908	Alex.C.Humphreys	L. F. Bayer	C. H. Colvin
Syracuse University	Dec. 3, 1911	W. E. Ninde	O. W. Sanderson	R. A. Sherwood
Univ. of Arkansas	Apr.12, 1910	B. N. Wilson	M. McGill	C. Bethel
Univ. of California	Feb.13, 1912	Joseph N. LeConte	J. F. Ball	G. H. Hagar
Univ. of Cincinnati	Nov. 9, 1909	J. T. Faig	A. O. Hurxthal	E. A. Oster
Univ. of Illinois	Nov. 9, 1909	W. F. M. Goss	A. H. Aagaard	H. E. Austin
Univ. of Kansas	Mar. 9, 1909	F. W. Sibley	E. A. Van Houten	L. E. Knerr
Univ. of Maine	Feb. 8, 1910	Arthur C. Jewett	E. H. Bigelow	O. H. Davis
Univ. of Missouri	Dec. 7, 1909	H. Wade Hibbard	W. P. Jesse	R. Runge
Univ. of Minnesota	May12, 1913			
Univ. of Nebraska	Dec. 7, 1909	J. D. Hoffman	A. A. Luebs	G. W. Nigh
Univ. of Wisconsin	Nov. 9, 1909	A. G. Christie	W. K. Fitch	J. W. Griswold
Washington University	Mar.10,1911	E. L. Ohle	D. Southerland	A. Schleiffarth
Yale University	Oct.11, 1910	L. P. Breckenridge	C. E. Booth	O. D. Covell

A NEW METHOD OF ELECTING MEMBERS

In order to provide against unnecessary delays in admitting to the Society those well qualified for membership, a new method of handling applications was adopted at the recent Spring Meeting.

Candidates for membership are posted in the issue of The Journal following the receipt of their applications. A period of 40 days is given in which members may advise the Secretary of any objection they may have to the election of any individual (see page 5).

At the expiration of the time for which an applicant is posted the Membership Committee meet to consider each application and make recommendation to the Council as to the grade to which candidates who receive their favorable consideration shall be assigned.

A list of the candidates, together with the recommendations of the Membership Committee, is then submitted to the Council for vote by letter ballot. Two negative votes prevent an election to membership.

This method of handling applications for membership makes it possible to pass upon well qualified candidates in from 70 to 90 days at any season of the year.

Members are requested to scrutinize with the utmost care the lists of applicants published in The Journal each month, in order that they may advise the Secretary of any candidate whose eligibility for membership is in any

way questioned. All correspondence in regard to such matters is strictly confidential.

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